

A COGAS PROPULSION CYCLE
WITH PEAK EFFICIENCY AT LOW POWER

Geoffrey Armstrong Clough

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Abstract

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This thesis is an engineering analysis of a proposed propulsion cycle for a destroyer type ship. It is organized and written in the same manner that an engineer would approach the problem. It starts with the requirements for a destroyer propulsion system and proceeds through a cycle selection process and individual component selection. At each step in the process the feasibility of each component is analyzed. After all the components have been selected, a heat balance is made to ensure that they all fit together. This is followed by a weight, volume, and reliability analysis. Much of the more detailed and voluminous material is included in appendices to retain continuity in reading the main body.

Thesis Supervisor: A. D. Carmichael
Professor of Power Engineering

Acknowledgements

Each person who helped in the research and development of this cycle and thesis is recognized at the appropriate spot in the text. But there are several men who gave much more aid than can be noted in this manner.

Mr. William H. VanCott, Marine Consultant to General Electric Company, conceived this cycle and initially spent many hours discussing the thesis with me. He suggested many alternatives at each step in the development of the thesis.

Mr. A. O. White, Manager of Marine Products of the General Electric Gas Turbine Department, spent two full working days with me at the onset getting the gas turbine portion started. In addition, he put me in touch with the proper men to get reliable information on heat recovery, combustion, etc.

Mr. David Gray and Mr. Carl Horlitz of Combustion Engineering spent a great deal of time discussing the boiler portion of the cycle. Additionally, Mr. Horlitz actually sat down and designed the boiler used and made the modifications necessary to use it in this cycle.

Mr. Chester W. Stott, Senior Engineer of The General Electric Steam Propulsion Group, spent many hours with me during the heat balance calculations. Without his knowledge and assistance the heat balance would have been very difficult indeed.

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Introduction

Traditionally, destroyer type ships are designed for the full power, maximum auxiliary load condition. Actually, these ships spend up to 90% of their underway time at speeds below 20 knots and much of this time is at speeds below 15 knots. A typical destroyer or modern destroyer escort seldom has an electric load exceeding 800KW and almost never exceeding 1000KW. When all the combat equipment is in operation, the electric load is at the rated capacity, but only then.

Using this argument, destroyer type ships should be designed for peak efficiency at a power level corresponding to about 20 knots with an electric load sufficient to cover the normal cruising requirements. In the case of destroyer escorts ships, this cruise power should correspond to the speed most commonly used in convoy or escort duties. Non-classified information shows this power level to be about 4500 SHP for a single screw DE at 15 knots and about 6000 SHP per shaft for the twin screw DD at 20 knots.

Specific fuel rate is not the only factor to consider in naval ship propulsion. Some other important factors are: volume, weight, reliability, maintainability, availability and response time to reach full power from various conditions. Each propulsion system is best in one or more of these areas and no propulsion cycle is best in all. For example, the gas turbine is by far the best when one considers response time and availability. The aircraft gas turbine is by far the best if one is concerned about weight and volume of propelling machinery, but this is offset somewhat if one examines weight and volume of the entire machinery package and cruise fuel load. The diesel engine is by far the most economical

user of fuel, but it is very large and noisy at the power levels necessary for a destroyer.

The combination of all these factors have led the naval engineer to look at combined power plants in order to achieve the optimum for his design. Some of the combinations considered have been:

Gas Turbine and Diesel (CODOG, CODAG)

Gas Turbine and Gas Turbine (COGAG, COGOG)

Gas Turbine and Steam (COGAS)

Proponants of each of these systems have expounded at length on the virtues of each in the literature. But to date no one has studied the combination proposed in this thesis.

This combination consists of a relatively small gas turbine for base (cruise) load using waste heat recovery. There is a relatively large steam boost to the full power requirements. At the cruise mode the boiler is unfired and the steam pressure low. At full power the boiler is fully fired and the steam pressure high.

All the components of this particular combination are available off-the-shelf at this time or are easily designed and manufactured. None of the equipment used in this cycle is beyond the current state of the art. Suggested improvements are made at each step in the process. Some would greatly improve the cycle if they were possible now.

At first glance this cycle looks complicated and confusing. But a deeper analysis shows that it is quite simple and could be made even simpler as the state of the art improves.

Procedure

The procedure used in the development of this thesis is much like that used by any marine engineer in the development of a concept. Once the basic idea for the cycle was formulated, components had to be found which were suited to it. This process included many inquiries made to individual experts in each of the fields involved. In some cases actual visits to design locations and manufacturing sites were necessary in order to provide a good flow of information both ways. In other cases telephone calls were sufficient.

In the development of a propulsion cycle there are several components which are very difficult to design and require extremely long lead times. In these cases it is mandatory to use existing equipment and to design the rest of the plant around them. This cycle had to be designed around an existing gas turbine and had to use a steam turbine which was only slightly modified from existing design. The boiler design is a modification of an existing design.

After the major components were selected and sized, a determination had to be made on electric power generation and auxiliary equipment. These were sized to fit the cycle and the decision made on the source of power to each.

Once all the components were sized, a heat balance was made. This step required making a few modifications to the cycle before a satisfactory result was obtained. Then a weight, volume and reliability study was made to provide some comparative numbers to use if this plant is to be compared with others.

Alternative suggestions are discussed at each step in the process and future state of the art suggestions are discussed in these steps if appropriate there. If not, they are left for the recommendations and discussion sections.

Results

Speed Profile

In a ship propulsion cycle, the first step is to determine the power level of the design points. In order to do this, the speed profile of the ship has to be known or assumed. Figure 1 shows the speed profile which was assumed for this study. It is based on assumption using the following reasoning.

The speeds from 0 to 10 knots are assumed as one block because the power level is so flat at these speeds for a destroyer type ship. There is a peak at the most efficient speed which is typically 14 or 15 knots since this can be assumed to be the speed most used in an independent transit. There is another peak at about 24 to 25 knots as this is currently about the maximum sustained speed using two boilers on a destroyer type ship. The rapid fall-off above 25 knots merely shows that these ships seldom operate in this speed range. The reasoning for this fall-off is that the power requirement increases so rapidly at these speeds that it would be uneconomical to run there unless operational requirements made it unavoidable.

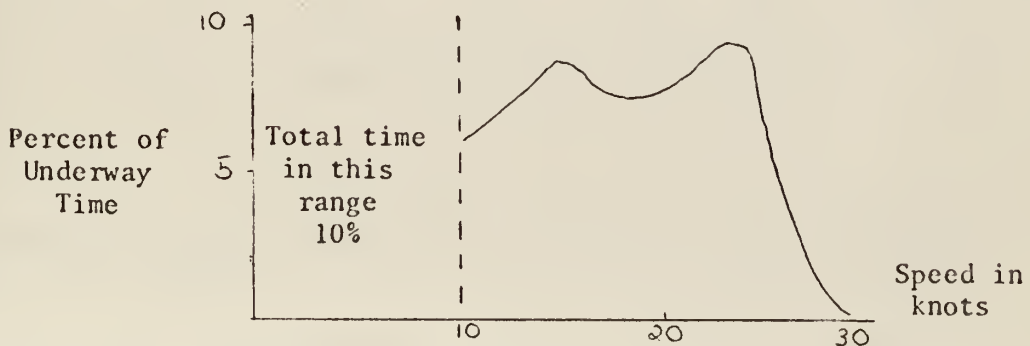


Figure 1. Speed Profile

Power Requirements

Having an assumed speed profile, it is now necessary to find the power levels at these speeds to determine the design power levels of the propulsion cycle. Figure 2 was compiled using turbine and gear instruction books for applicable plants.

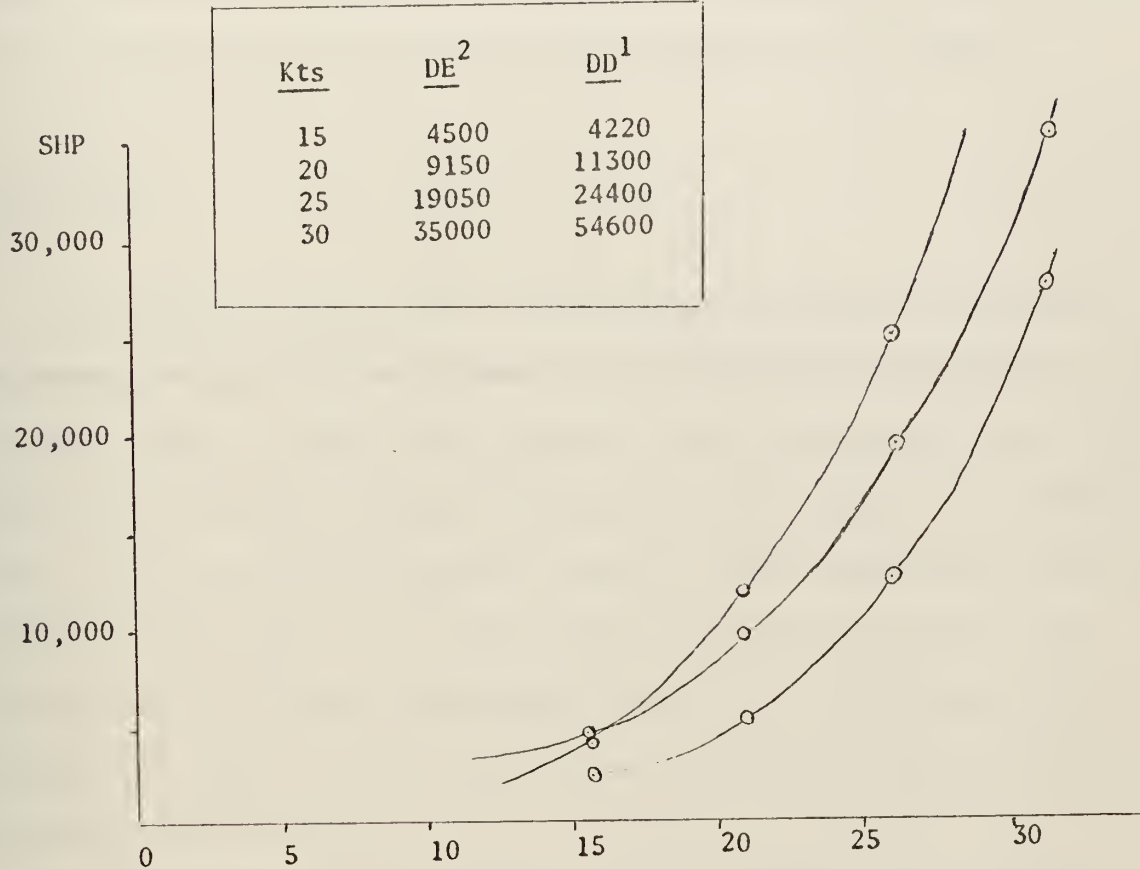


Figure 2. Power Verses Speed in Knots Curve

¹Turbine and Gears Instruction Book for the DD - 936, (NavShips 341-1301, GEI - 56786).

²Turbine and Gears Instruction Book for the DE/DEG, (NavShips 0941-011-8010, GEI - 60465).

These curves show that a power plant for a destroyer type ship should reach its peak efficiency at about 9000 SHP corresponding roughly to 20 knots in modern destroyer hulls. The peak power level should be about 35,000 SHP to be competitive with existing plants.

The electric power requirements were set at 1000KW for the normal underway load with an installed capability of 4000KW in at least two separate units for redundancy. This decision is discussed in the electric power section but is mentioned here for continuity of discussion.

Cycle Choice

The propulsion cycles open to choice after satisfying the above requirements are very limited. There is the conventional steam plant currently used by the navy with a specific fuel rate of about 0.65 at this power level or the more advanced marine plants with specific fuel rates of about 0.44 if it has five feedwater heaters. A more complicated plant than the five heater plant is the reheat steam plant where the steam is extracted from the turbine, reheated and piped back to the turbine again. This plant is capable of fuel rates down to 0.41 but is too complicated to be feasible for navy use.

Another choice is the straight gas turbine engine. Until recently, this cycle has suffered from a poor fuel rate. Currently the most economical of these is the second generation LM-2500 aircraft engine manufactured by General Electric and slated for use in the DD 963 class of destroyers and the PF patrol boats. This engine has a very admirable fuel rate at full power of 0.39 (propulsive). At 9000 SHP this becomes 0.51 (propulsive);

and in order to compare it with the cycle for this thesis, it has to be corrected to an all purpose fuel rate. This figure then becomes about 0.60 when corrected for electric power and evaporator requirements. This figure is a little better than the current navy plants but worse than the merchant ship plants. Even more disquieting is what happens when two of these engines are run at the 20 knot power level which is what can be expected in the currently planned installations except when economy is the overriding consideration. Here the propulsive fuel rate becomes about 0.71 and the all purpose fuel rate about 0.91. A current navy plant runs about 0.82 at this power level and the merchant ship plant is considerably better.³ These figures are summarized in graph form in appendix A.

Another alternative cycle is the combined diesel and gas turbine. This plant is hard to find fault with other than it requires some form of clutching with the associated problems of large clutches. Diesel engines of the size required are also large and have a problem with low frequency noise. This cycle is currently in use on the U. S. Coast Guard Hamilton class cutters.

Still another alternative cycle is the combined gas turbine and gas turbine cycle. One turbine would be small for cruise power and one large for full power. This system allows the gas turbine to operate at its best level most of the time eliminating the fuel rate problems. The clutches can also be eliminated by accepting the windmilling losses of the idle turbine. This cycle has also been used in some installations.

³Comparative All Purpose Fuel Rates Chart, prepared by the ship propulsion section of General Electric Company, MST division, January 1971.

By far the most interesting and challenging cycle considered for this application is the combined steam and gas turbine cycle. This cycle has been manufactured for many years for land use and has been labeled a STAG cycle (Steam Turbine and Gas Turbine). In each installation to date this cycle has used the steam plant for base load with a large gas turbine for peaking power. This particular combination offers no advantage for a marine power plant as it would be easier to provide more or larger boilers to begin with if more power were desired.

A variation of the STAG cycle is a gas turbine with a heat recovery steam generator. This plant uses the heat which would be wasted in the gas turbine exhaust to generate steam which can be used for propulsion, hotel services or auxiliary loads. This plant, too, has been manufactured for many years both with fired and unfired boilers.

A marine application of this waste heat recovery cycle was the GTS JOHN SARGENT which was placed in service in 1956 and run for some 9700 hours. In this application the gas turbine provided all the propulsion power and the steam was used for powering auxiliaries and for providing hotel services. There was no supplemental firing.⁴ Several other marine applications of this cycle are being designed but none are using supplemental firing.

The cycle analyzed in this thesis is one ideally suited to the destroyer application. There is a gas turbine for cruising supplemented

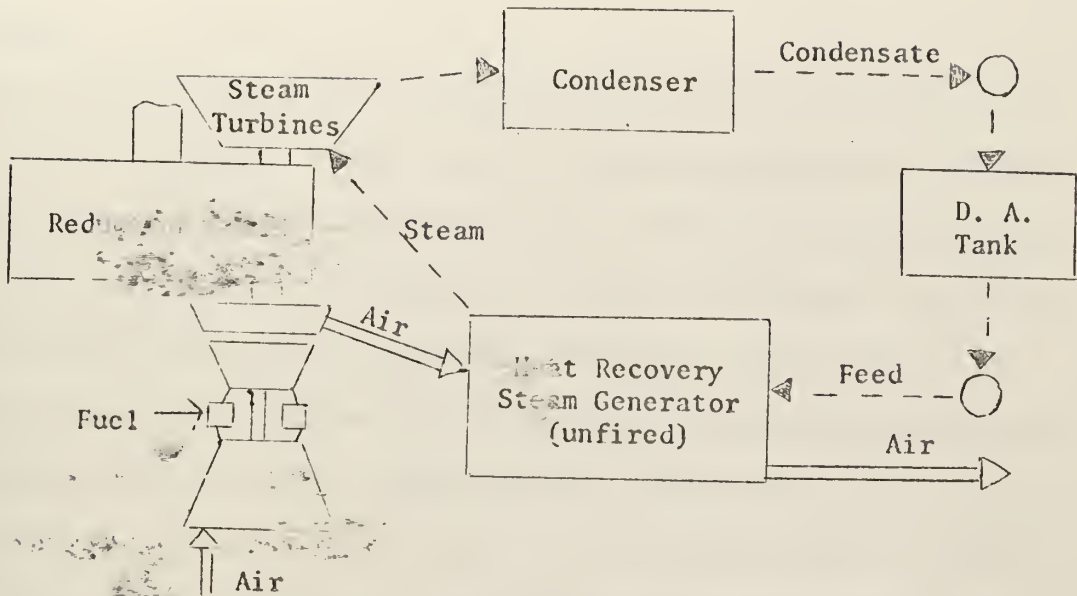
⁴Personal discussions with William H. VanCott, Marine Consultant under contract to General Electric Company (Lynn, Mass., June 1971 - August 1971). Mr. VanCott's prior experience includes almost 30 years at sea including duty as first engineer of the SS United States and chief engineer of the GTS John Sargent.

by a steam turbine using heat recovered from the gas turbine exhaust. In the cruise configuration there is no supplemental firing of the heat recovery steam generator. This is one design point for the boiler and where it should have its highest efficiency. As the power requirements increase, the boiler is fired as required. Ideally one would use a grid type burner and heat the exhaust gas from the gas turbine just before it passes through the boiler. The analysis in appendix F shows the power attainable under these conditions. The current state of the art of grid burners does not allow for this. They fire well only using natural gas and foul using heavy distillate or residual fuels. Therefore, the boiler in this study had to use a forced draft blower and conventional burners. This question is addressed again in the boiler section.

This cycle uses many simplification features and many efficiency improvers such as: variable steam pressure, controllable and reversible propeller, variable geometry gas turbine, constant inlet area steam turbine and an attached generator. It also has many reliability improvements, the greatest of which is the boiler. It is always hot as there is always hot gas passing through it. When it is fired, it is fired by a distillate fuel which virtually eliminates fireside maintenance. Figure 3 shows the cycle diagram at low and high power level in greatly simplified form.

Up to this point only the basic cycle has been outlined. Now the component selection must be made. Once again it must be stated that these components are not always the ideal ones but are those which come closest to ideal in keeping with the stated objective of using existing equipment or, at worst, readily redesigned and manufactured equipment. Further optimization is discussed in each section.

Cruise Power Level



Full Power Level

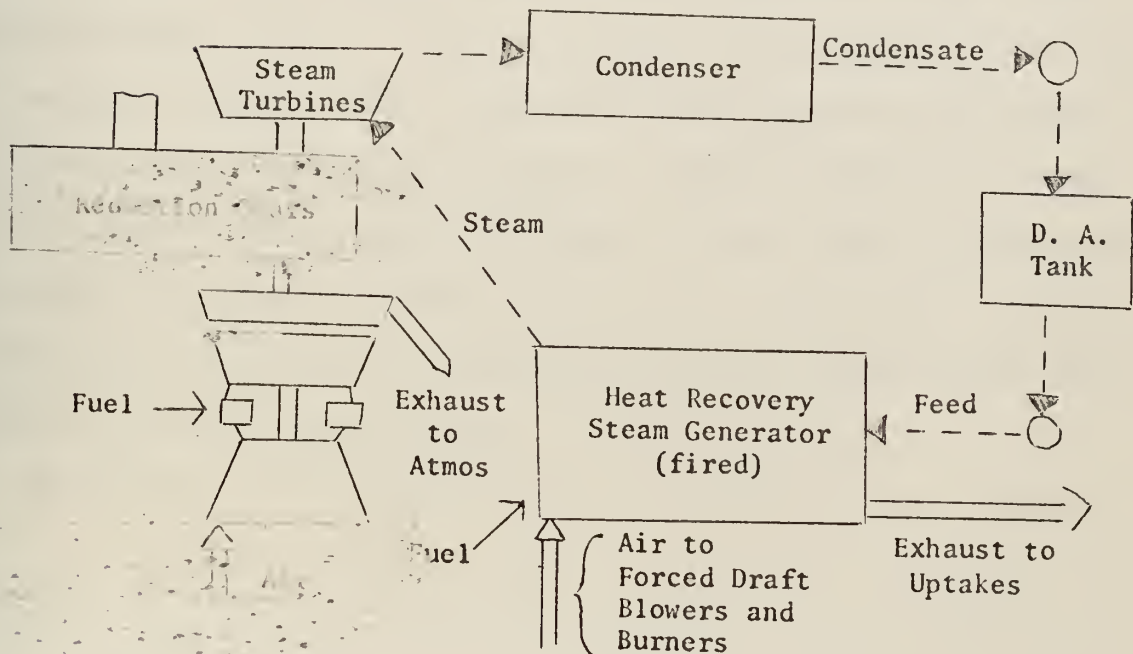


Figure 3. Cruise and Full Power Cycle Configurations

Component Selection

Gas Turbine

The single most complicated and difficult to design component in the cycle is the gas turbine. There are many turbines available off-the-shelf designed for many different applications with the majority being designed for aircraft use. All the aircraft gas turbines are designed with frontal area and weight minimization of paramount importance. Only the second generation LM-2500 has combined this weight and volume minimization with an outstandingly good fuel rate. This particular gas turbine has two disadvantages for use in this cycle. First, it has too much power to operate efficiently at the required power level, and secondly, it has the typical overhaul cycle discussed below.

All aircraft gas turbines have two common severe limitations when applied to marine plants. These are the short time between overhauls and the poor part load fuel rate. The overhaul cycle problems arise because all these turbines operate at the limits of current technology in temperature and pressure ratios in order to reduce size and weight. Consequently the overhaul interval is typically 10,000 hours of operation.⁵ The part load efficiency problems can be partially overcome by using variable area geometry at the power turbine inlet.

By contrast, the overhaul interval of the industrial gas turbines is

⁵J. V. Shannon et al., "DX Engineering Plant Life Cycle Cost Comparison Basic Steam Plant Vs. Basic Gas Turbine Plant Arrangement (GT - 1), Marine Turbine and Gear Dept., General Electric Company (Lynn, Mass., 16 October 1968).

typically 1000,000 hours of operation.⁶ This is primarily due to a more conservative design approach using much less critical parameters and much less exotic metals.⁷ One does pay a severe penalty in machinery size and weight because of this, but these industrial gas turbines still sell because the size and weight does not matter in a land plant installation.

In a marine plant one would ideally want a turbine designed for long hours of continuous use with an overhaul as long as or longer than that for the rest of the ship. This can be satisfied only by an industrial gas turbine. But, in a marine plant one also wants minimum space and quite often minimum weight as well. This dictates an aircraft gas turbine. The trade off on which way to go depends on the type of ship the installation is for. In a supertanker neither weight or space is very critical. In a container ship space is critical but weight is not. In a naval destroyer both are very critical which dictates using an aircraft gas turbine unless there is another way out.

An industrial gas turbine which is small enough to fit into the general size and weight limits imposed on a naval destroyer has been selected for this plant. This particular turbine also gives adequate power for cruise when waste heat recovery is employed. The turbine is the General Electric series 1000 which is currently designed for about 4900 SHP output. Some of the design parameters are listed in Table 1 which is

⁶G. A. Ludwig, "Marinization of Industrial Heavy-Duty Gas Turbines," General Electric Gas Turbine State of the Art Engineering Seminar (SOA-17-71), June, 1971.

⁷A. O. White, "General Electric Heavy-Duty Gas Turbines," General Electric Gas Turbine State of the Art Engineering Seminar (SOA-16-71), June, 1971.

extracted from references 6 and 9, parts of which are included in appendix B.

<u>General Electric Model 1000 Gas Turbine</u>			
Number of Stages:	Compressor		15
	Turbine		2
Shaft Speed:	Gas Generator	10920 rpm	
	Power Turbine	10920 rpm	
Aux. Power Req.	A.C. Lube Oil Pump	15 HP	
	D.C. Emerg. L. O. Pump	5 HP	
	A.C. Ignition	500 watts	
	D.C. Control	125 watts	
	Starter, 450 VAC	100 HP	
Rotation:	CW when facing load end		
Performance on Distillate Fuel:	Air	161,700	Lb/Hr
	s.f.c.	0.5668	Lb/SHP-Hr
Weight on foundation with accessories:		18600	Lbs

Table 1.

In addition to this information, various conversations with Mr. A. O. White, Manager of Marine Products of the General Electric Gas Turbine Department, indicated the following possible improvements. This turbine could have its power output and air flow increased by 20% by a modification called "zero staging." This consists of adding one more stage of compressor blading ahead of the existing blading changing the output to that listed in Table 2. The fuel rate would remain about the same.

SHP	6000	
Air Flow	194000	Lb/Hr

Table 2.

Mr. White also indicated that a substantial weight reduction would be possible for a naval design, but this reduction is not used at this time since it would require a substantial redesign and since firm estimates of the weight reduction were not possible.

This turbine is well suited to this application if "zero staged." Due to the subsequent requirement of using forced draft blowers at full power, one could use this turbine even without the modification because the extra performance and gas flow is not required. This would decrease the overall plant output by 20% which would increase the specific fuel rate about 30%. In addition to being well suited to use in this cycle, all the performance data is available at design and for off design and all the dimensions and weights are available. This eliminates a great deal of estimating and makes a determination of feasibility more believable.

Steam Turbine

The steam turbine design for this cycle includes some features not usually included in a steam turbine because of the need to match it to the gas turbine. It must run at a constant speed over its entire power range. This is because the gas turbine is limited to a 2% overspeed to its alarm point and only 10% overspeed to its failure point.⁸

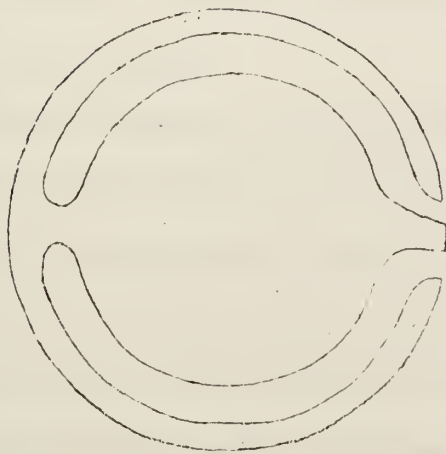
⁸ A telephone conversation with Mr. A. O. White, Manager, Marine Projects for the General Electric Company Gas Turbine Division, 9 July 1971.

The heat balance in appendix F shows that the power level at the cruise design point is about 3200 SHP. This is the point where the gas turbine is at full power and the boiler is unfired. From this point, the steam turbine must go to its maximum output of about 36,000 SHP.

Current technology limits on metal, pumps, etc. limit steam pressure to about 1450 psig. Steam plants of higher pressure are being considered but none have been installed in marine plants. The pressure at full power in this cycle was set at 1200 psig. This was mainly because that is the pressure currently used by the U. S. Navy. The temperature was similiarly limited to 950°F. Since this temperature cannot be reached in the unfired mode because of limitations on heat exchanger size, the steam temperature must vary. This raised the question of looking at variable steam pressure also.

Variable steam pressure is very well suited to the steam turbine design and allows a constant area inlet for most of the power range. The steam pressure was allowed to vary from 400 psig at the cruise mode to 1200 psig when at full power. These pressures were compatable with the boiler design and allow sufficient pressure at low powers to run all the required auxiliary equipment. The low limit on the pressure was not completely compatable with the steam turbine design and required the inlet area to be halved at the cruise mode. The basic idea is shown in Figure 4. This still eliminates the need for the complicated inlet nozzles present in all standard steam turbines. This modification is only a casing modification and can be easily accomplished. The power output of the turbine is controlled by the modulating valve until it is wide open at 400 psig. Thereafter the power is controlled by varying the steam pressure. If a

faster response is required, a full 1200 psig could be carried on the boiler and the power controlled by the modulating valve over the entire range, but this sacrifices efficiency.



This inlet controlled by a modulating valve.

This inlet controlled by an open or shut valve.

Figure 4. H. P. Steam Turbine Inlet Arrangement

The choice of operating temperature was based on the boiler design where a temperature of about 750°F at 400 psig and 950°F at 1200 psig was attainable from the same boiler. This is mentioned here because these temperatures are necessary to estimate the turbine efficiency.

The steam turbine efficiency was estimated using established procedures and then backing out the corrections for astern loss (there is no astern element), speed loss (constant speed) and part load loss (variable pressure).⁹ The steam turbine has only exhaust loss when using constant inlet area, constant speed and variable pressure. This is a very strong

⁹"Marine Steam Power Plant State of the Art Seminar, 1970," Babcock & Wilcox and General Electric Company (eds.).

argument for this cycle configuration. The estimate for steam turbine efficiency is shown in appendix C and is 0.790 at the cruise point and 0.824 at the full power point. It should be noted here that these figures are not as accurate as shown, but they are accurate to within about 2%. This is because this turbine has not been analyzed in sufficient detail to ensure more accuracy. However, this accuracy is more than adequate for this feasibility analysis.

The decision of whether to use a single cylinder or a multi cylinder turbine is primarily a function of the exhaust pressure and the steam rate combined with the room available for the turbines. In other words, it is a function of how much exhaust loss is acceptable. In order to keep the condenser size and weight down, a decision was made to accept 5 inches of mercury pressure at full power. This was justified by the fact that at cruise power this would be 1.1 inches and there would be little lost efficiency. A further justification was the little time actually spent at the higher power levels. This decision forced the use of a cross-compound turbine which had already been recommended because of the power level.¹⁰

Heat Recovery Steam Generator

Preliminary calculations on the steam flow possible were made using procedures established by Sheldon and Todd and are shown in appendix F.¹¹ Extended talks with Mr. Sheldon in late July, 1971, showed that the cycle

¹⁰Conversation with Mr. M. A. Prohl, Manager, Turbine Engineering, Marine Turbine and Gear Department, General Electric Company, August, 1971.

¹¹R. C. Sheldon and D. M. Todd, "Optimization of the Gas Turbine Exhaust Heat Recovery System," ASME Paper 71-GT-79.

was possible although the feasibility of a boiler for it was questionable. Further research in this area proved that a convective heat transfer boiler could not satisfy both the cruise and full power requirements. Therefore, the boiler has to use radiant heat transfer at full power.

It should be pointed out here that boiler design is not a science but more of an art. No one to date has been able to explain exactly what happens inside a boiler, but they can come close in boilers of a conventional design. When the design is like the one for this cycle where there is no radiant heat transfer in one mode and there is in the other mode, the calculations must be made by hand. These calculations require the empirical formulas and charts developed over many years of boiler design. Even then the calculations are an iterative process.

The boiler design calculations are included in appendix D. These calculations were made by Mr. Carl Horlitz of Combustion Engineering, Inc. and by the author of this thesis. They are accurate to within 10% which is sufficient for the purposes of this feasibility study. This, in fact, is about the degree of accuracy attainable in a boiler design of this type. A more accurate study can only be made on a boiler of conventional design where the parameters are fairly well known to each design.

The boiler selected for this cycle is the Combustion Engineering, Inc. boiler designed for use in the AOE class ships for the U. S. Navy. The reasons for this choice were many. First, it is all ready designed to navy standards and this cycle is applicable to navy ships. Second, it has the required full power flow rate. It is easily upgraded to the 1200 psig required for full power. It was then necessary to calculate whether or not this boiler would fulfill the cruise requirements.

As can be seen in appendix D, the boiler does fit the cruise requirements, but would require a slight modification to the economizer. It also required a modification to the air flow path. The air would enter through the floor of the boiler rather than the front in the cruise mode. This modification was made to reduce the pressure drop through the boiler and retain the maximum gas turbine performance possible.

Another change had to be made from the originally conceived design in that the gas turbine exhaust has to bypass the boiler at full power. This is due to the high pressure losses in the conventional burners and the fact that there simply is not sufficient oxygen available in the exhaust gas to fire the boiler to the required full power level. If grid burners (burners consisting of a patchwork of pipes with holes on the downstream side which provide a large but low density flame area) could be made to burn heavy distillate fuel well, this type of burner could be placed in the gas path at the bottom of the boiler. This would allow a steam rate which would be considerably below the 230,000 Lb/Hr desired. But this type of firing would be sufficient for almost all the underway speed requirements of a destroyer type ship as shown in the assumed speed profile. If grid burners were satisfactory, they are what should be installed. One could then use a supplemental forced draft blower and burner for the final boost.

This requirement to bypass the gas turbine exhaust in the fired mode adds considerable complexity to the system in the form of additional ducting, a bypass valve, and much larger forced draft blowers than originally conceived. This is an unfortunate necessity forced upon the cycle because of the state of the art of grid burners.

The results of the boiler calculations are summarized in Table 3. The full power steam temperature is just a little too high and should be

lowered to 950°F. This adjustment was made for the heat balance calculations. This is a minor adjustment and could be made by removing some of the superheater surface area or using an attemperator which is the current practice in merchant plants where a constant superheater outlet temperature is desired. The heat balance also showed that some desuperheated steam was required in the cruise mode. This adjustment was also made during heat balance calculations.

Summary of Boiler Calculations

	<u>Cruise Power</u>		<u>Full Power</u>	
	<u>Blr Calc</u>	<u>Heat Bal</u>	<u>Blr Calc</u>	<u>Heat Bal</u>
Air: Firebox	950°F	- -	2752°F	- -
After Superheater	816°F	- -	1757°F	- -
After Main Bank	492°F	- -	761°F	- -
Stack	364°F	- -	355°F	- -
Steam and Water:				
To Economizer	280°F	280°F	280°F	280°F
	415 psig	415 psig	1275 psig	1275 psig
To Main Bank	448°F	- -	407°F	- -
	415 psig	- -	1275 psig	- -
In Steam Drum	448°F	- -	578°F	- -
	415 psig	- -	1275 psig	- -
Superheater Outlet	758°F	758°F	997°F	955°F
	400 psig	400 psig	1200 psig	1200 psig
Desuperheater Outlet	- -	456°F	655°F	655°F
	- -	395 psig	1160 psig	1160 psig
Flows: Air (Lb/Hr)	194000	194000	293238	293238
Superheated Stm (Lb/Hr)	25000	22250	230000	210000
Desuperheated Stm (Lb/Hr)	0	4370	30000	31600
Draft Losses (In. Water)	4.36	- -	48	- -

Table 3.

The extremely low draft loss at the cruise power is noteworthy. This loss allows the gas turbine to operate at very near its design point in this mode. At full power the draft loss is considerably higher and reflects the burner losses rather than any great increase in the boiler loss. The draft loss through the boiler if the burner loss is neglected is about 15 inches of water.

Figure 5 is a diagram of the air flow arrangement of the boiler. The gas bypass and the reasoning for the floor located air inlet has been explained. The ducting immediately below the boiler must be brick lined as shown because of the intense heat in the firebox when the boiler is being fired.

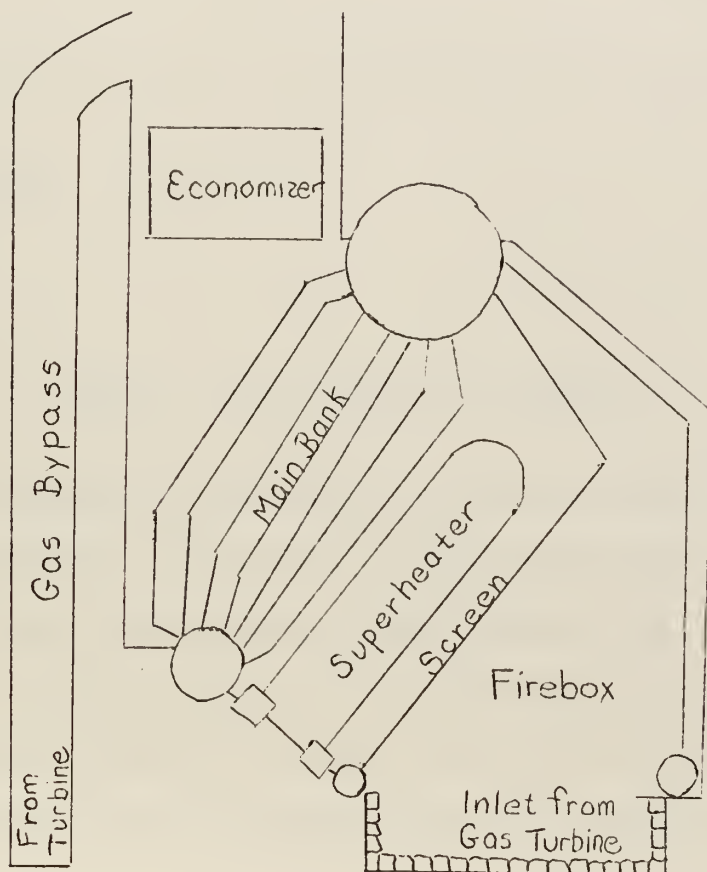


Figure 5. Boiler Arrangement Diagram (Front View)

Figure 6 shows a schematic of the ducting for the exhaust gas routing system. This damper is arranged to minimize possible damage to the gas turbine due to inadvertant shutting or flareback. The two dampers would be connected mechanically to ensure that one or the other is always open. They are hinged at the upstream end so that the gas flow would keep them open should the mechanism fail.

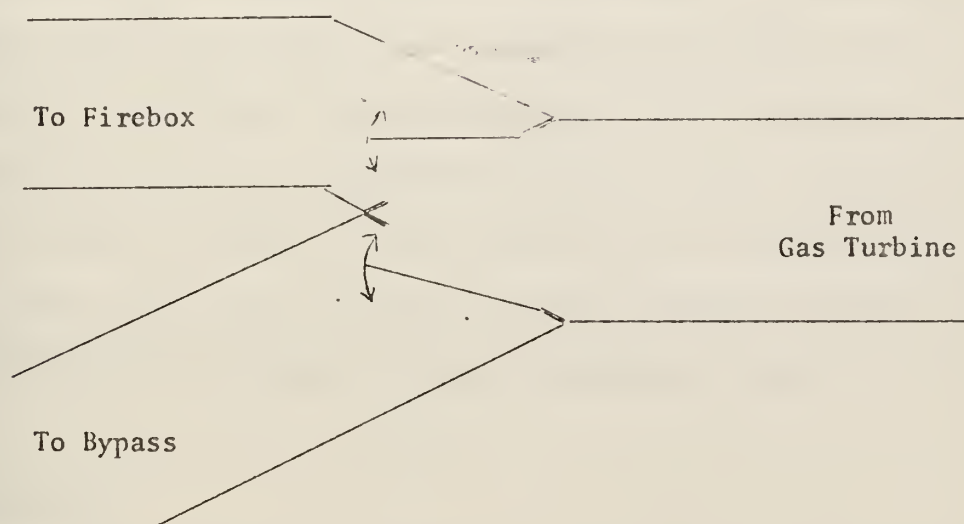


Figure 6. Air Flow Ducting Schematic

The boiler bypass could be routed to either the inlet or the outlet side of the economizer. Because the full power mode where there would be a full bypass of gas turbine exhaust gases would be used so seldom, the total effect of routing these gases to either location would be minimal. Therefore, they were routed to the outlet side of the economizer to keep exhaust losses down on the gas turbine and to keep the total height of the boiler down.

Using the configuration of this proposal, the reaction time to raise

the boiler from the cruise condition to the full power condition would be on the order of 7 to 8 minutes. If a faster response time were needed, the controls could be arranged to keep the boiler at 1200 psig at all times. This would reduce the reaction time to about 2 to 3 minutes. But it would also reduce the cruising efficiency of the plant by requiring a less efficient steam turbine. It would also reduce the plant maintainability by requiring much more stringent feedwater requirements. It would also reduce the boiler reliability somewhat due to continued operation at the higher temperatures and pressures. The reaction time of 7 to 8 minutes does not seem unduely long when one considers the number of times that maximum acceleration would be desired. There could even be an emergency mode on the control system which would allow this constant pressure operation when desired since the steam turbine does have a modulating valve.

Reduction Gears

The reduction gears chosen for the cycle are a straight forward, naval designed, double reduction, locked train type much like those installed at present. These gears allow the spread required to fit in the cross-compound steam turbine and the gas turbine. This required width ruled out planetary gears. Figure 7 shows a typical set of reduction gears for a destroyer type ship. The set required for this cycle would add an input on the after end for the high pressure steam turbine and a take-off for the attached generator (discussed later).

Gugliuzza and Hargett give sample drawings of several gear arrangements one of which is for a COGAS cycle designed to fit into a DD-931 class destroyer. This particular cycle used five input shafts with an articulated

gear train for all inputs instead of the conventional locked train type. The reason for this departure from normal was due to the number of gears they had to cluster around the bull gear. The cycle proposed here needs only two outputs or inputs on each end and, therefore, should easily fit a conventional design. Further discussion by Smith on the state of the art of gear design leaves little doubt that this is possible.¹²

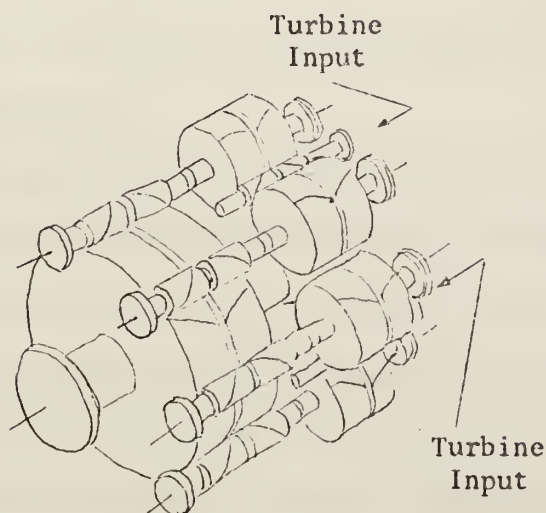


Figure 7. Typical Destroyer Gear Set¹³

Propeller

The choice of the proper propeller for this cycle is restricted to controllable and reversible pitch propellers because of the constant speed requirement of the gas turbine over the entire upper range of power. At

¹²"Marine Steam Power Plant State of the Art Seminar, 1970," Babcock and Wilcox and General Electric Company (eds.).

¹³T. A. Gugliuzza and W. H. Hargett, "Gear Design and Laboratory Test Experience--Marine Turbine Propulsion," ASME Paper 69 - GT3.

the lower power levels of the gas turbine, such as for maneuvering, more power is available faster using the constant speed control, so the C & RP propeller is still better.

The controllable and reversible propeller offers a flexibility unmatched by any fixed pitch propeller and can be programmed to give whatever pitch one desires at any power level and rpm. This leaves the final choice of control sequence open to any variations required by the particular application of the cycle. The curves included in appendix E show just how versatile these propellers are.

The decision on controlling the propeller for this cycle was made after a discussion with the Bird-Johnson Company of Walpole, Massachusetts. At the power levels of this cycle there could be a strong cavitation signature if the choice is a constant speed shaft, but this type of control is entirely possible. In this same discussion it was indicated that this signature problem should be overcome in the not-too-distant future. Since constant speed offers several efficiency advantages, the decision was made to use it. The two major improvements of constant speed on the shaft are: a greatly improved steam turbine efficiency at part load and the ability to use an attached generator for normal cruising electric power. The turbine efficiency improvement factor can be seen in appendix C and is about 0.85 at half speed. The improvement by using an attached generator is discussed in the following section.

Appendix E has a line drawing of a BLH controllable pitch propeller designed for use in naval destroyers of about this horsepower. It shows the hub and control unit mechanics. Also included in this appendix are typical control sequences for this propeller.

There is much controversy over whether one loses or gains efficiency with controllable pitch propellers. The further one pursues this question, the more it becomes obvious that the answer depends on who is giving the opinion. One school of thought insists that the larger hub and general purpose design of the individual blades cause a loss of propulsive efficiency. The other school of thought is that, while there is some loss of efficiency at the design point, the C & RP propeller is more efficient at off design and reverse points. Therefore, for a naval destroyer, which operates over a wide range of speeds continuously, the C & RP propeller appears to have a significant advantage. In addition, it allows for much faster response in a maneuvering situation.

Electric Power Generation

The last major decision to be made prior to conducting a heat balance is that of electric power generation. The choices here consist of steam, gas turbine, or diesel powered or attached generators. In recent studies, the diesel generator has been ruled out because of its low frequency noise and maintenance problems.

By far the most efficient method of generating electric power is by means of an attached generator. For this method the main shaft must run at a fixed speed. The generator would be run off the reduction gear and, therefore, could not be very large because of space limitations in this cycle.

The next most efficient method is to use a steam turbine as the power source since there is already steam present in large enough quantities. The turbine could exhaust into the main condenser thereby using the greater

vacuum attainable there over an auxiliary condenser.

The third choice would be for a gas turbine power source. Here again all the pros and cons associated with industrial verses aircraft turbines arise. Since they were discussed previously, they will not be discussed here.

Because efficiency is one of the main goals of this cycle, the decision was made to go with an attached generator large enough to carry the normal underway electric load. This will make it small enough to fit aft of the reduction gear beside the main shaft. The size of this generator was set at 500 KW on each shaft.

Naval destroyers often require much greater powers when running all the combat equipment. But these requirements are only for short durations under normal conditions although they could be for extended time in combat. Therefore, the decision was to use an industrial gas turbine power source. These generators would be instantly available for peaking loads and could also be run for extended times when required due to the industrial design.

The particular choice for this cycle was the AVCO TF25A Industrial design gas turbine which will run on navy distillate fuel. This engine develops about 1500 KW continuously and 1650 KW maximum at a specific fuel rate of about 0.693 Lb/SHP-Hr or 0.924 Lb/KW-Hr. These are the numbers used in the heat balance at full power where additional power is required to run the electric pumps in the propulsion plant.

The 1500 KW power level was chosen to place the total installed electric power capability at 4000 KW. In a two shaft ship using two of these generators plus two attached generators this number is achieved.

There could be a problem with synchronization of the gas turbine and

attached generators if the ship were maneuvering violently. Therefore, in the maneuvering situation one would probably use the gas turbine generators.

Auxiliaries

The various auxiliaries for this cycle were sized using the references listed in each case and integrating them with established guidelines.¹⁴

All the specific pressures and steam flows are in the heat balance.

Main Condenser: This was designed by standard methods of reference 4. This item was sized in order to attain 5 inches of mercury absolute pressure at full power and then the vacuum at cruise power was calculated.

Forced Draft Blowers: Reference 7 was used to find the required steam flow at full power.

Feed Pumps: Reference 10 was used to establish the power requirements of the pumps. The decision was made to use a multiple speed electric motor for the power because of the variable pressure of the output and the greater efficiency of the electric motor over a steam pump.

Circulating Water Pump: This pump is an electric driven pump which is installed but not used underway above approximately 5 knots. It is, therefore, not included in the heat balance calculations.

Heat Balance

The heat balance results are in appendix F. These results show that

¹⁴S. T. Holm, "Recommended Practices for Preparing Marine Steam Power Plant Heat Balances," (revised in 1970 by C. W. Stott, Jr), SNAME Technical and Research Bulletin, No. 3-11a.

the cycle does in fact fit together. The fuel rate is not as good as originally expected or attainable with modifications. But the components for the cycle were chosen with a naval application in mind and, therefore, some efficiency was sacrificed. Examples of these losses are an unusually large evaporator load, a greater than normal electric load, a lower than normal feedwater temperature, a cycle with only two feedwater heaters, and relatively inefficient forced draft fans.

	<u>Sheldon & Todd</u>	<u>Heat Balance</u>
Steam Turbine Horsepower	3100	3180
Gas Turbine Horsepower	<u>5760</u>	<u>5760</u>
Total Horsepower	8860 SHP	8940 SHP

Table 4.

Weight Analysis

A summary of weights for this plant is shown in Table 5. In each case the reference or references are listed. Some degree of comparison can be made with the General Electric proposal for the DX project which had an installed equipment weight of 686,000 pounds. Standard guidelines for steam plant installations allow for much more than that.

When a propulsion plant weight comparison is being made, the total weight to be compared should include everything necessary to accomplish the same task. The electric loads should be comparable in that the power available to the ship after the propulsion plant power is subtracted should be

Summary of Propulsion Plant Installed

Equipment Weights

	<u>WT. (Lbs)</u>	<u>References</u>
Main Propulsion Turbines and Gears	148000	15
Propulsion Gas Turbine with Accessories	16000	9
Lube Oil System (Cooler, Pumps, Strainers, Purifier, Heater)	12000	15
Feed Water System (Cruise Pump, Main Pump, Booster Pumps, DFT)	42000	15
Vacuum and Condensate System (Condenser, Vacuum Pump, Circulating Water Pump, Condensate Pumps, Gland Leakoff Cond., Gland Exhaust Fan, Drain Tank and Pump)	88000	15
Steam Generation System (Boiler, Controls, Fans)	230000	15, 2
Electric Power Generation (Attached Gen., Gas Turbine Generator)	28000	1
Miscellaneous	<u>14000</u>	15
TOTAL	578000 Lbs	

Table 5.

the same for each case. The evaporator load should be adjusted so that each case considered has the same number of gallons per man available after the propulsion plant requirements are met. A weight comparison should also include the weight of the fuel required to meet endurance specifications. The figures used to make the fuel calculation should be the all purpose fuel rate in each case. Table 6 shows a sample comparison of the fuel load to meet endurance. These figures are for a 7755 SHP per shaft power level

using attached generators at the 500 KW load. The electric load of the steam plant is met by steam powered generators at the 500 KW load.

If a lower power level were compared, the gas turbine plant would increase its fuel consumption dramatically and the steam plant would increase to a lesser degree. The cycle analyzed in this thesis would increase only slightly.

<u>Comparison of Cruise Fuel Load for Three</u> <u>Comparable Propulsion Plants</u>		
This Cycle (One shaft)	0.438	333 tons
1200 psi DD Plant (One shaft) [Current U.S.Navy practice]	0.64	497 tons
LM-2500 (One operating, evaps incl.)	0.57	437 tons

Table 6.

Volume Analysis

It is clear that the gas turbine plant has a considerable advantage over other propulsion plants as far as propulsion equipment is concerned. This advantage is reduced somewhat when the total plant volume is considered. However, the cycle proposed in this thesis does have an advantage over a conventional steam plant.

There is no second boiler. Even though the one boiler is large when

compared to typical destroyer boilers, it is small when compared to the two required in the steam plant installations. The total fireroom length would be reduced by about 8 feet in the DE-1052 Class. The engineroom, however, would have to be about the same length.

The auxiliary equipment space would be about the same volume as that in a conventional steam plant but would be reduced somewhat by the absence of auxiliary condensers and their associated equipment. The gas passage ducting could raise this again to have the volume about the same as that of a steam plant. This ducting would be routed down from the gas turbine and along the hull to the bottom of the boiler. In this manner the volume would be used where is normally wasted space in a conventional steam plant.

Reliability

The reliability study was made using Prantis's publication as a guide.¹⁵ The items not included in this reference were estimated using information made available by the manufacturer in some cases and simply guessed at in some other cases. These values are shown in Table 7. The mission time was taken at 225 hours or about the total endurance time of this plant. The full power reliability is: $R = 0.983$. The cruise power reliability is: $R = 0.984$. The "take home" power reliability is $R = 0.999$.

A part of reliability is redundancy. This is one area where this cycle is very good. The ship would be able to run on either the gas turbine or on

¹⁵Edwin R. Prantis, "Tanker Steam Plant Reliability," Paper presented to the New England Section of SNAME, 15 January 1971.

the boiler alone. In a casualty situation they could be uncoupled and the remaining one used.

<u>Reliability Data</u>	
	$\lambda \times 10^6$ (Failures per Million Hrs $\times 10^6$)
Boiler	11.0
Forced Draft Blower	6.0
Steam Turbine	5.3
Main Condenser	3.0
Main Condensate Pump	6.0
Main Vacuum Pump	6.3
Heat Exchanger and Accessories	4.0
Main Lube Oil Pump	4.5
Main Lube Oil Cooler	2.0
Fuel Oil Service Pump	7.0
Deaerating Feed Water Heater	1.0
Main Feed Booster Pump	7.0
Main Feed Pump	8.0
Gas Turbine	14.0
Adjustable Damper	1.0
Ship's Service Electrical System	17.2
Main Reduction Gear	0.7
Shaft, Bearings & C&RP Propeller	2.6

Table 7..

Discussion of Results

This thesis describes a feasibility study of a particular ship propulsion cycle. Each of the components was chosen using existing or easily modified and manufactured equipment. There are several areas where the efficiency and desirability of this cycle could be improved if the state of the art allowed. This is particularly true in the case of the grid burner and in the case of the auxiliary equipment as discussed in the heat balance section. In the case of the auxiliary equipment, further efficiency improvements could be made by accepting merchant practices rather than naval practices.

The weight and volume analysis is very dependent on the specific application of the cycle and the specific choices of various components. Therefore, no attempt was made to be any more accurate than necessary to show that these parameters are competitive with existing cycles.

One of the great advantages of this plant not discussed in the text is the ease of maintenance of the plant as a whole. The gas turbine should require very little maintenance other than the normal overhaul. The boiler spends almost all of its life cycle at pressures from 400 psi to 600 psi and is designed for 1200 psi operation. This should make boiler waterside maintenance minimal. The fireside maintenance should be similarly reduced because the boiler is seldom fired; and when it is, it is fired with distillate fuel which virtually eliminates fireside deposits. The feedwater quality tolerances should be set for 1200 psi operation and, therefore, should further reduce waterside problems.

Conclusions

This thesis has shown that this cycle is feasible which was the primary goal. Further, it is feasible to assemble the cycle using existing or readily available equipment.

The desirability of the cycle is much less obvious and requires some thought on the part of the reader after the first introduction to it. The desirability of the extremely good fuel rate is obvious. But the question of plant complexity appears to overpower the good fuel rate until some thought is given to it. The most complex question in the cycle is that of control. This study has assumed a constant speed propeller for reasons given in the text. There are many components in the cycle which have to be controlled separately and integrated to give smooth transition from cruise to full power.

The control of the cycle would be much easier if a grid burner were included as there would be very little mechanical manipulation to get from cruise power to about 70% power. Above this power level, there would have to be a boiler bypass in order to get the firebox hot enough. But up to that point the only control necessary would be to change propeller pitch and increase the fuel to the boiler. Both of these could be related directly to steam drum pressure.

The cycle appears to be quite complex at first glance. But, if one looks long enough at it, he can see that the complexity of the steam portion has been reduced greatly. The gas turbine is not a very complex machine, but is almost completely self contained and can be programmed to be controlled in a great variety of ways. In the case of this cycle, it would

remain at full speed as the power is reduced.

When considered as a whole cycle and all the advantages and disadvantages are considered, it appears that this cycle is worth considering for a prototype installation and for further study.

Recommendations

This cycle should be applied to a very specific use and a complete study accomplished including the control question. Then a prototype plant should be assembled and run.

Should this cycle ever be considered for an actual shipboard installation, the engine should be designed and sized to that particular installation and every step should be taken to optimize the cruise fuel rate at the power level required for cruise. This study has taken the maximum cruise rate possible for its optimization.

References

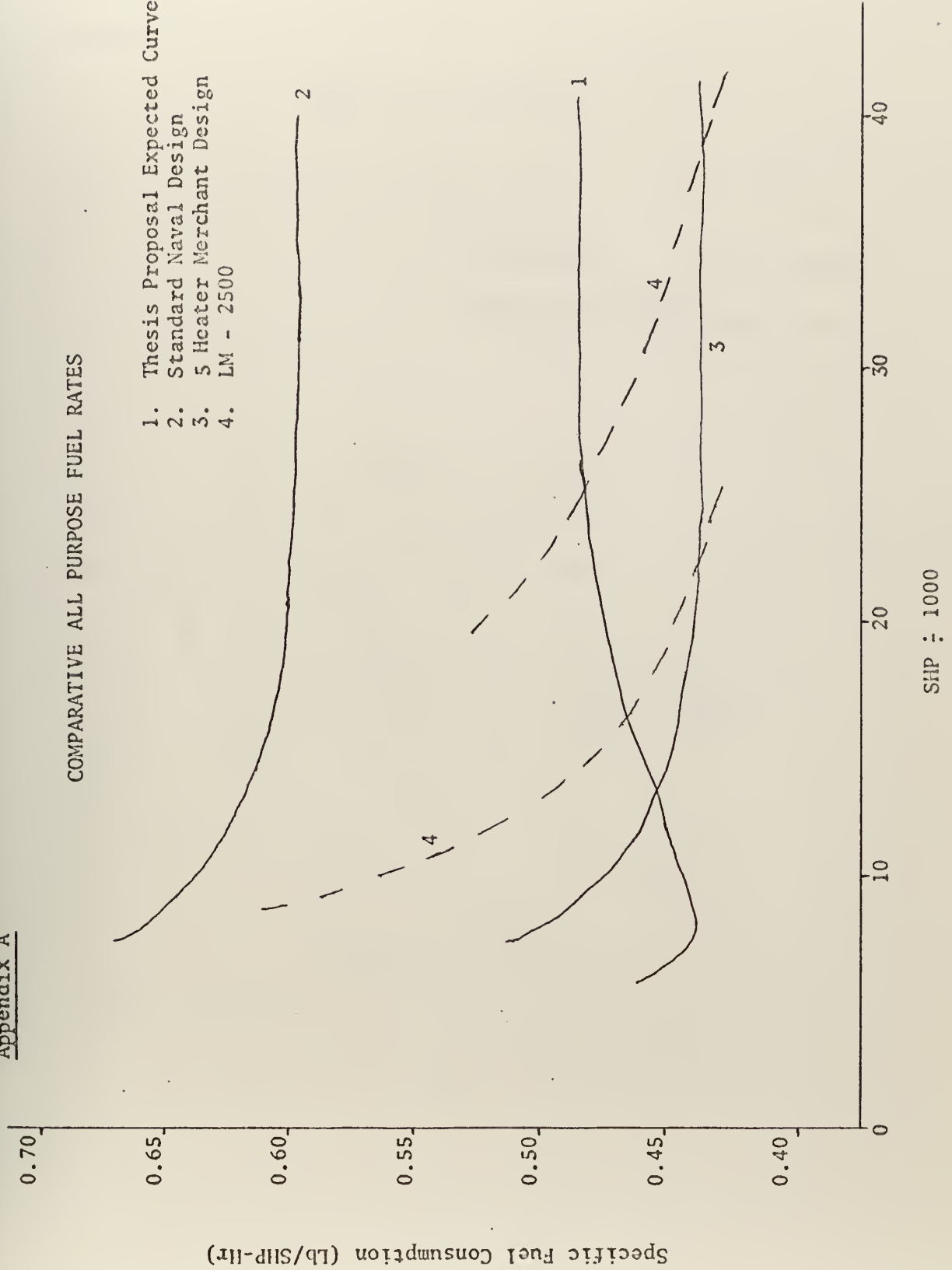
1. Application Handbook; Turbine-Gears and Turbine-Generators for Ship Propulsion, Turbine Generators for Ships Service, U.S. Navy Applications, General Electric Company, 1965.
2. Combustion Engineering Drawing, F-167-431-C Main Boiler, AOE-3.
3. "Comparative All Purpose Fuel Rates Chart," prepared by the ship propulsion section of General Electric Co., MST division, January, 1971.
4. Condenser, C-H-Wheeler Mfg. Co., Philadelphia, Pa.
5. "Four Quadrant Open Water Test for Controllable-Pitch Propeller 924.1," Hydronautic, Inc., Technical Report 7082.10, December, 1970.
6. "General Electric Model 1502* 5050 HP Gas Turbine Design," Drawing No. 418 HA 604, 4 December 1970.
7. Giblon, R. P. et al, "Design Considerations for Boiler Forced Draft Systems," George G. Sharp, Inc., 1969.
8. Gugliuzza, T. A. and Hargett, W. H., "Gear Design and Laboratory Test Experience--Marine Turbine Propulsion," ASME Paper 69-GT3.
9. "Heavy-Duty Gas Turbine for Marine Propulsion," Model Series 1000, a sales information brochure (GEA-9307), put out by General Electric Company, Gas Turbine Dept., Schenectady, New York.
10. Holm, S. T., "Recommended Practices for Preparing Marine Steam Power Plant Heat Balances," revised in 1970 by C. W. Stott, Jr., SNAME Technical and Research Bulletin No. 3-11a.
11. Kapsenberg, I. C., "Controllable Pitch Propellers, Their Suitability and Economy for Large Seagoing Ships Propelled by Conventional Directly-Coupled Engines," June, 1964.

12. Ludwig, G. A., "Marinization of Industrial Heavy-Duty Gas Turbines," General Electric Gas Turbine State of the Art Engineering Seminar (SOA-17-71), June, 1971.
13. "Marine Steam Power Plant State of the Art Seminar, 1970," Babcock & Wilcox and General Electric Company (eds.).
14. Prantis, Edwin R., "Tanker Steam Plant Reliability," paper presented to the New England section of SNAME, 15 January 1971.
15. "A Proposed Steam Turbine Propulsion Plant for the DX Project," General Electric Company, Steam Propulsion Group, Lynn, Mass.
16. Shannon, J. V. et al, "DX Engineering Plant Life Cycle Cost Comparison Basic Steam Plant vs. Basic Gas Turbine Plant Arrangement (GT-1)," Marine Turbine and Gear Dept., General Electric Co., Lynn, Mass. 16 October 1968.
17. Sheldon, R. C. and Todd, D. M., "Optimization of the Gas Turbine Exhaust Heat Recovery System," ASME Paper 71-GT-79.
18. Standards for Closed Feedwater Heaters, Heat Exchange Institute, New York, 1968.
19. "35000 Horsepower Controllable Pitch Ship Propeller," Description and Assembly Instructions, Baldwin-Leuice-Hamilton (BLH) Company.
20. Turbine and Gears Instruction Book for the DD-936, NavShips 341-1301, GEI-56786.
21. Turbine and Gears Instruction Book for the DE/DEG, NavShips 0941-011-8010, GEI-60465.
22. White, A. O., "General Electric Heavy-Duty Gas Turbines," General Electric Gas Turbine State of the Art Engineering Seminar (SOA-17-71) June, 1971.

Appendix A

COMPARATIVE ALL PURPOSE FUEL RATES

1. Thesis Proposal Expected Curve
2. Standard Naval Design
3. 5 Heater Merchant Design
4. LM - 2500



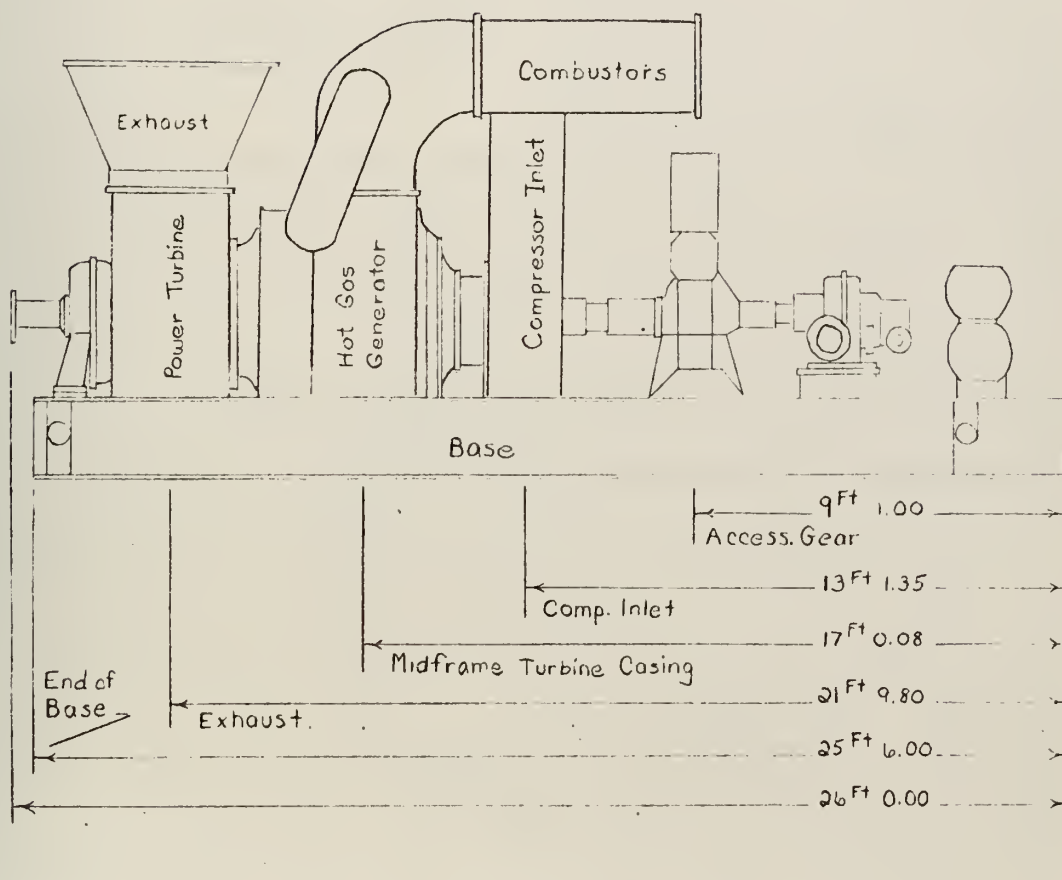
Appendix B

MODEL 1000 ELEVATION VIEW

Max. Width 10 FT 8 IN

Height to Top
of Combustor 11 FT 6 IN (approx)

Weights: Gas Turbine
with Base and Accessories 18,600 Lbs



GENERAL ELECTRIC MODEL M1502 *5050HP GAS TURBINE

ESTIMATED PERFORMANCE

COMPRESSOR INLET TEMPERATURE 59°F (15°C)

COMPRESSOR INLET PRESSURE 14.7 PSIA (760 mm of Hg)

FUEL

* NATURAL GAS

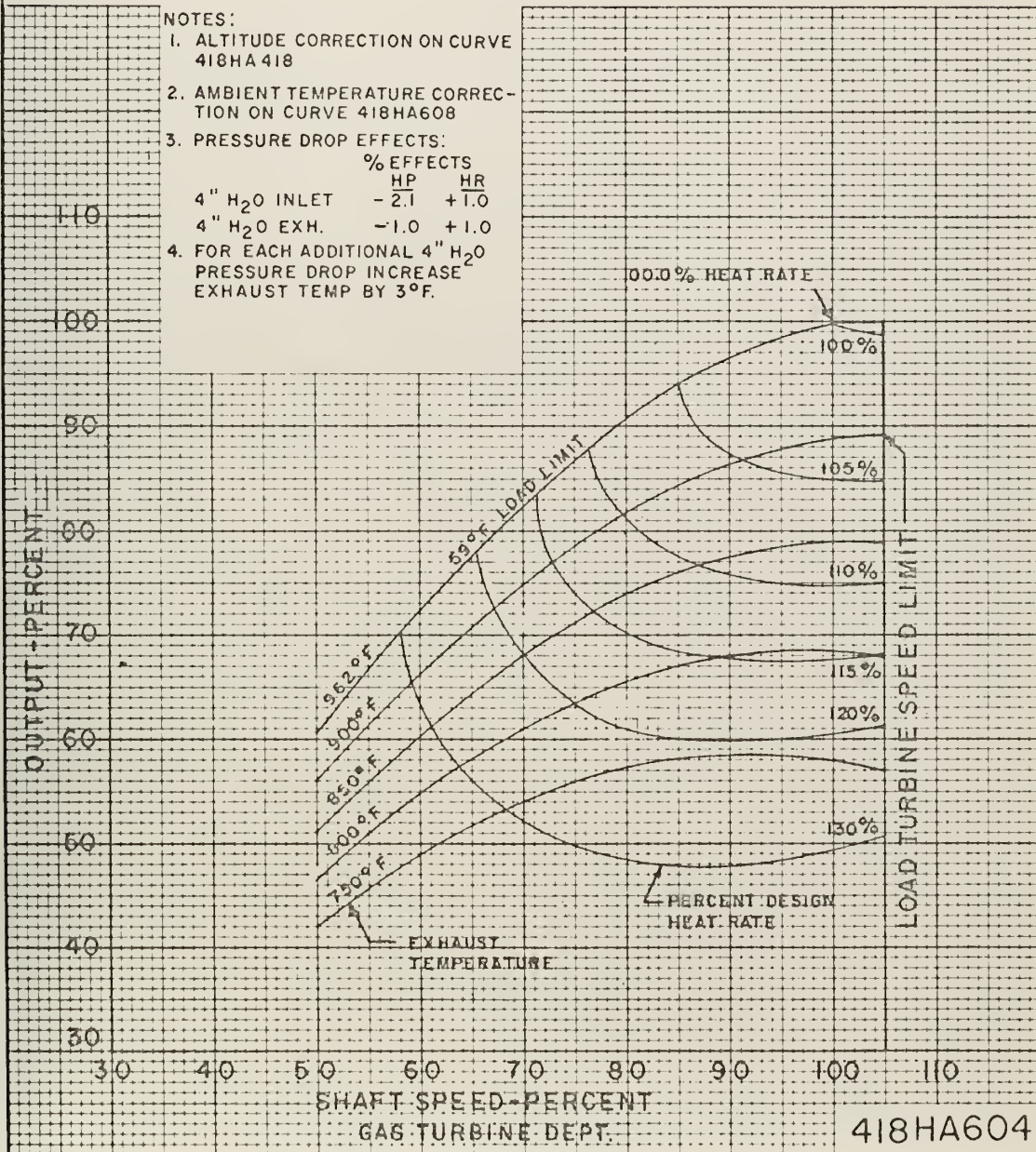
DISTILLATE OIL

DESIGN OUTPUT	HP	5050	4900
DESIGN HEAT RATE (LHV)	BTU/HP-HR	10200	10430
DESIGN FUEL CONSUMPTION (LHV)	BTU/HR	51.5×10^6	51.1×10^6
RATIO HHV/LHV		1.11	1.06
DESIGN AIR FLOW	161,700 LBS/HR		
DESIGN SHAFT SPEED	10290 RPM		

NOTES:

1. ALTITUDE CORRECTION ON CURVE 418HA 418
2. AMBIENT TEMPERATURE CORRECTION ON CURVE 418HA608
3. PRESSURE DROP EFFECTS:

	HP	HR
4" H ₂ O INLET	-2.1	+1.0
4" H ₂ O EXH.	-1.0	+1.0
4. FOR EACH ADDITIONAL 4" H₂O PRESSURE DROP INCREASE EXHAUST TEMP BY 3°F.

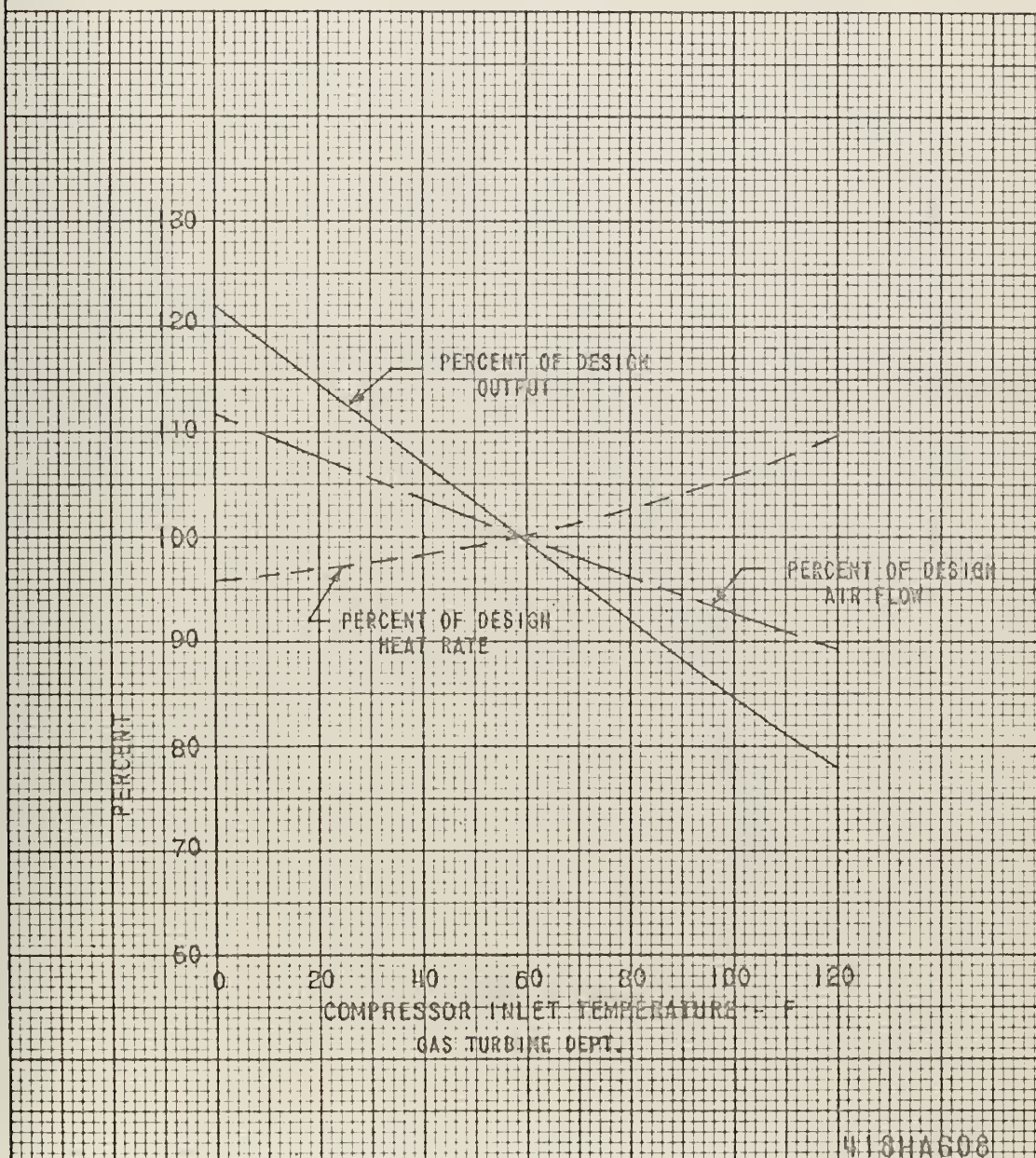


GENERAL ELECTRIC MODEL SERIES 1502 GAS TURBINE

EFFECT OF COMPRESSOR INLET TEMPERATURE ON
MAXIMUM OUTPUT, HEAT RATE, AND AIR FLOW

100% SPEED

Curves for: 100% Compressor Speed
1675°F Firing Temperature



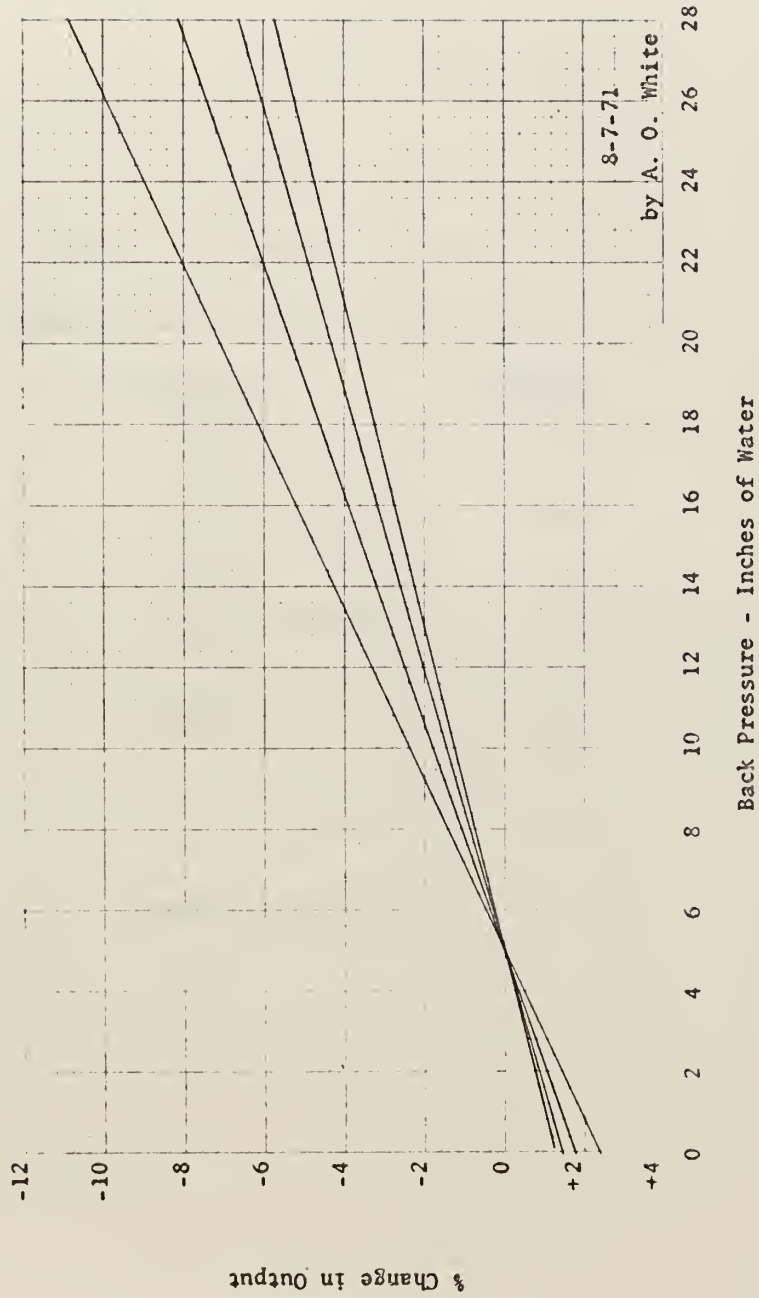
413HA608

Appendix B

GAS TURBINE (Two-Shaft Cycle)

BACK PRESSURE CORRECTION

AT 70°F AMBIENT



Appendix B

ANALYSIS OF INLET AIR

	<u>By Wt</u>	
O ₂	23%	C _p = 0.24 Btu/Lb
N ₂	77%	60% Relative Humidity 80°F

ANALYSIS OF EXHAUST AIR USING

DISTILLATE OIL AT F/A RATIO 0.01308

	<u>By Volume</u>	<u>By Weight</u>
CO ₂	2.6%	4.1%
H ₂ O	4.7%	3.0%
O ₂	16.2%	18.1%
N ₂	76.5%	74.9%
SO ₂	0.0034%	0.0076%

ANALYSIS OF FUEL OIL

API Gravity	33°
Diesel Index	45.5
C/H Ratio	6.7
Sulphur Content	0.3%
LHV	18,350 Btu/Lb

Extracted from: General Electric Gas Turbine Department PEI 4.1.1
12-10-62

Appendix C

ESTIMATION OF STEAM TURBINE PERFORMANCE

(based on references 10 and 13)

	<u>Full Power</u>	<u>Cruise</u>
Basic Turbine Efficiency (ref. 10)	0.84	0.84*
Temperature Correction (ref. 10)	1.02	0.99
Exhaust Loss Factor (ref. 13)**	0.95	0.96
Estimated Turbine Efficiency	0.824	0.790

*Reference 10 gives excellent "good practice" rules for calculating the efficiency but must be combined with what the author was told by Mr. Prohl.

**The calculation procedure in reference 13 is quite involved and complicated and must be corrected for various changes one would make for this cycle.

Steam Rates

Single-Cylinder or Cross-Compound

SCOPE OF PERFORMANCE DATA

The performance tables in this section include combinations which could result in impractical designs. However, in order to include all of the combinations which it might be desirable to study, the scope selected is necessary.

PROCEDURE

Steam rates at rated steam conditions and rated and partial shaft horsepower (shp) are to be calculated as follows:

- Read theoretical steam rate in lb per hp-hr from Theoretical Steam Rate Tables, 1936 Edition, published by The American Society of Mechanical Engineers (Section 4707 of General Electric Company Apparatus Handbook).
 - Convert to lb per hp-hr by multiplying by 0.7455.
- Read basic efficiency from Table A1 or A2 at rated horsepower and initial pressure.
- Read initial temperature correction factor from Table B.
- Obtain load correction factor from Table C1 or C2 at speed function = $\sqrt{\text{power function}}$.
- Correct basic efficiency (b.) \times (c.) \times (d.).
- Obtain astern rotation loss factor from Table D.
- Calculate method-exhaust flow:

$$\text{M.E.F.} = \frac{\text{Power} \times \text{T.S.R.}}{(\text{e.}) \times (\text{f.}) \times 0.98}$$

- Determine minimum annulus:

$$(\text{g.}) \div 6000 \text{ Back Pressure Inches Hg. A.}$$
- Determine maximum annulus:

$$(\text{g.}) \div 4000 \text{ Back Pressure Inches Hg. A.}$$
- Select annulus to be used. (See Note 4, Table E.)
- Determine flow factor:

$$(\text{g.}) \div [\text{BP} \times (\text{j.})]$$
- Obtain excess exhaust loss from Table E.
- Calculate exhaust-loss factor:

$$(\text{m.}) = 1.00 - \left(\frac{1}{100} \right) \left(\frac{(\text{i.})}{26.5} \right) \left(\frac{(\text{a.})}{(\text{e.})} \right)$$

- Calculate engine efficiency:

$$(\text{n.}) = (\text{e.}) \times (\text{f.}) \times (\text{m.}) \quad 72.2\%$$
- Calculate nonextracting steam rate:

$$(\text{o.}) = (\text{a.}) \div (\text{n.})$$

Example:

See page 52 for example calculations.

Tolerance on Shipboard Performance Tests

Steam rate guarantees on all turbines are made on the basis of compliance being demonstrated by precision test as specified in the ASME Turbine Test Code PTC6-1964. The instrumentation required by this code cannot be obtained aboard ship. Therefore, when compliance with guarantees is demonstrated by test aboard ship a tolerance for additional testing error is required:

- If the ship has a torsionmeter and calibrated shaft installed, both approved by the Medium Steam Turbine

Generator and Gear Dept. of the General Electric Co., and has properly calibrated throttle pressure, throttle temperature and condenser vacuum gages, and calibrated condensate flow meters, the main propulsion turbine shall meet the guaranteed steam rate after corrections for deviations in throttle pressure, throttle temperature, condenser vacuum and propeller RPM with a 1 percent tolerance for shipboard testing accuracy.

- If the ship has an approved torsionmeter installed but a standard shaft modulus has been specified in lieu of a calibrated shaft, and has properly calibrated throttle pressure, throttle temperature, and condenser vacuum gages, and calibrated condensate flow meters, the main turbine shall meet the guaranteed steam rate after corrections for deviations in throttle pressure, throttle temperature, condenser vacuum and propeller rpm with a 2% tolerance for shipboard testing accuracy. In either case the test should endure for one hour with steady-state readings on the prime variables.

Allowable Tolerance on Operating Variables

Initial Steam Pressure

For any given load, the steam pressure shall not average more than that specified. In maintaining this average, the pressure shall not exceed 110 percent of that specified. During abnormal conditions, the pressure may swing momentarily to 120 percent of that specified, but the aggregate of such swings shall not exceed 1 percent of the total specified operating life. (3.4.5.1.2.1 MIL-T-17600B.)

Initial Steam Temperature

For any given load, the steam temperature shall not average more than that specified. In maintaining this average, the temperature shall not exceed the specified temperature plus 15 F, except that during abnormal conditions the temperature shall not exceed, (a) the specified temperature plus 25 F for not more than 5 percent of the total specified operating life, or (b) the specified temperature plus 50 F for swings of 15 minutes duration or less, aggregating not more than 1 percent of the total specified operating life. (3.4.5.1.2.2 MIL-T-17600B.)

Back Pressure

The back pressure at the turbine exhaust flange may vary from a minimum of 0 inches Hg to a maximum of 1 1/2 inches Hg greater than the specified back pressure.

Valves Wide Open Throttle Flow

At specified steam conditions, the turbine is designed to pass a valves wide open throttle flow 5 percent greater than that required to develop the maximum specified power and provide the specified extraction flows.

Extraction Flow

Extraction openings will be sized to pass the specified extraction flows with nominal pressure drops. Extraction flows of up to 15 percent of the throttle flow may be extracted from a given opening with increased pressure drop.

Speed

The propeller speed for a given horsepower may vary ± 3 percent from the specified speed power curve for the ship. During abnormal conditions, the speed for a given power may be decreased by 30 percent provided that the maximum

specified SHP is not exceeded and provided that such operation does not aggregate more than 2 percent of the total specified operating life.

NUMERICAL EXAMPLE OF METHOD

20,000 Shaft hp - 575 Psig - 840 FTT - 1.75 Inches Hg. Abs.

Entity	Units	Source	Example Calculation			
1. Power	Shaft hp	Given	6860	10240	16000	20000
2. Power Fraction	—	Given Shaft hp Max. Shaft hp	0.343	0.512	0.80	1.00
3. Speed Fraction	—	$\sqrt[3]{\frac{1}{0.343}}$.70	.80	.928	1.00
4. Initial Pressure	PSIG	Given	625	625	600	575
5. Initial Temperature	$^{\circ}$ FTT	Given	860	860	850	840
6. Back Pressure	Inches Hg. Abs.	Given	1.00	1.25	1.50	1.75
7. Theoretical Steam Rate	Lb. hp-hr	Lbs. per Kw-hr (from T.S.R. Table) $\times 0.7455$	4.615	4.700	4.830	4.947
8. Base Efficiency	—	Table A (P_1 Max. Shaft hp)	.8248	.8248	.8260	.8268
9. Temperature-correction Factor	—	Table B	1.001	1.001	1.000	.999
10. Load-correction Factor	—	Table C	.8540	.9230	.9854	1.00
11. Corrected Base Efficiency	—	$(E_1)(E_2)(E_3)$.7051	.7620	.8139	.8260
12. Factor for Astern Rotation Loss	—	Table D	.997	.996	.995	.994
13. Method Exhaust Flow (M.E.F.)	Lb/hr	$\frac{(P_1)(T_1)}{(P_2)(T_2)} \times .98$	46000	64700	97400	123000
14. Minimum Annulus	ft ²	$\frac{Q}{32.2 \times 6000} \times 14.7$	—	—	—	11.71
15. Maximum Annulus	ft ²	$\frac{Q}{32.2 \times 4000} \times 14.7$	—	—	—	17.57
16. Annulus Used	ft ²	Note 4, Table E	—	—	—	13.65
17. $\frac{F}{\text{Inches Hg. Abs.}}$	Lb/hr/Inches Hg./ft ²	$\frac{Q}{A \times 14.7}$	3370	3790	4760	5150
18. Excess Exhaust Loss	BTU/lb	Table E	5.78	7.30	11.5	13.6
19. Exhaust-loss Factor	—	$1.00 - \left(\frac{1}{100} \right) \left(\frac{60}{26.5} \right) \left(\frac{1}{10} \right)$.9857	.9830	.9743	.9693
20. Engine Efficiency	—	$(E_1)(E_2)(E_3)(E_4)(E_5)(E_6)$.6929	.7460	.7890	.7958
21. Locus Best Point Non-extorting Steam Rate	Lb/hp-hr	$\frac{1}{(E_1)(E_2)}$	6.66	6.30	6.12	6.22

Steam Rates

Single-Cylinder or Cross-Compound

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TABLE A1

Basic Efficiency for Single-cylinder Main Propulsion Turbines (Interpolate for Intermediate Pressures and Ratings)

Max SHP	Steam Pressure (lbf Turbine Inlet, Psig)								
	200	300	400	500	600	700	800	900	1000
2000	0.737	0.726	0.718	0.710	0.702	0.694	0.687	0.682	0.675
2500	.758	.749	.741	.733	.726	.719	.712	.706	.700
3000	.774	.766	.757	.750	.742	.734	.726	.724	.717
3500	.783	.775	.767	.760	.752	.744	.736	.734	.728
4000	.791	.784	.776	.770	.762	.754	.746	.745	.739
4500	.797	.790	.782	.775	.767	.759	.751	.754	.748
5000	.801	.794	.787	.781	.773	.765	.757	.758	.752
6000	.805	.799	.793	.787	.780	.772	.764	.761	.754
7000	.810	.804	.799	.794	.788	.783	.778	.772	.768
8000	.814	.809	.804	.798	.793	.788	.783	.777	.773
9000	.818	.813	.808	.802	.797	.792	.787	.782	.778
10000	.822	.816	.811	.806	.801	.796	.792	.787	.783

TABLE B

Initial Temperature Correction Factors (Interpolate for Intermediate Pressures and Temperatures)

Initial Temp °F	Steam Pressure (lbf Turbine Inlet, Psig)				
	200	400	600	900	1500
500	0.948	0.942	0.936	0.931	
510	.950	.944	.938	.933	
520	.952	.946	.940	.936	
530	.953	.948	.943	.938	
540	.956	.949	.945	.941	
550	.958	.951	.948	.943	
560	.960	.954	.950	.946	
570	.962	.956	.952	.948	
580	.964	.958	.953	.950	
590	.965	.960	.955	.953	
600	.967	.962	.958	.954	0.949
610	.969	.964	.960	.956	.951
620	.970	.966	.962	.958	.954
630	.972	.967	.964	.960	.956
640	.974	.969	.966	.962	.958
650	.975	.971	.968	.964	.960
660	.977	.973	.970	.966	.962
670	.978	.975	.971	.968	.965
680	.980	.976	.973	.970	.967
690	.981	.978	.975	.972	.969
700	.983	.980	.977	.974	.971
710	.984	.981	.979	.976	.973
720	.985	.982	.980	.978	.975
730	.986	.984	.982	.979	.977
740	.988	.985	.983	.981	.979
750	.989	.987	.985	.983	.981
760	.990	.989	.987	.985	.983
770	.991	.990	.988	.987	.985
780	.993	.991	.990	.988	.987
790	.994	.993	.991	.990	.989
800	.995	.994	.993	.992	.991
810	.996	.995	.994	.993	.993
820	.997	.997	.996	.995	.995
830	.998	.998	.997	.997	.996
840	.999	.999	.999	.998	.998
850	1.00	1.00	1.00	1.00	1.00
860	1.001	1.001	1.001	1.001	1.002
870	1.002	1.003	1.002	1.002	1.003
880	1.003	1.004	1.003	1.004	1.005
890	1.003	1.005	1.005	1.005	1.006
900	1.004	1.006	1.006	1.006	1.007
910		1.006	1.007	1.008	1.008
920		1.007	1.008	1.009	1.010
930		1.008	1.009	1.010	1.011
940		1.009	1.010	1.011	1.012
950		1.010	1.011	1.012	1.014
1000			1.015	1.017	1.020
1050			1.018	1.021	1.026

TABLE A2

Basic Efficiency for Cross-Compound Main Propulsion Turbines (Interpolate for Intermediate Pressures and Ratings)

Max SHP	Steam Pressure (lbf Turbine Inlet, Psig)									
	200	400	500	600	700	800	900	1000	1250	1500
7000	0.810	0.799	0.794	0.788	0.783	0.778	0.772	0.768	0.755	0.742
8000	.814	.804	.798	.793	.788	.783	.777	.773	.761	.749
9000	.818	.808	.802	.797	.792	.787	.782	.778	.767	.754
10000	.822	.811	.806	.801	.796	.792	.787	.783	.772	.760
11000	.825	.814	.809	.805	.800	.796	.790	.787	.776	.764
12000	.828	.817	.812	.808	.803	.799	.794	.790	.780	.761
14000	.832	.822	.817	.814	.809	.805	.800	.797	.787	.776
16000	.836	.826	.822	.819	.814	.810	.806	.802	.792	.783
18000	.840	.830	.826	.822	.818	.814	.810	.806	.798	.788
20000	.843	.833	.829	.826	.821	.818	.814	.810	.802	.793
22000		.835	.831	.828	.824	.821	.817	.813	.805	.797
24000		.838	.834	.831	.827	.824	.820	.816	.808	.800
26000		.840	.836	.833	.829	.826	.822	.819	.812	.803
28000		.841	.838	.835	.831	.828	.824	.821	.814	.806
30000		.843	.839	.836	.832	.829	.826	.823	.816	.808
35000					.840	.836	.833	.831	.827	.821
40000					.843	.839	.836	.834	.830	.825
50000					.846	.843	.840	.838	.835	.830
60000					.849	.846	.843	.841	.839	.834
70000					.851	.848	.846	.844	.842	.837

Steam Rates

June 19, 1967

Single-Cylinder or Cross-Compound

TABLE C1

Load Correction Factors

(Interpolate for Intermediate Speeds) (Fraction of Speed = $\sqrt{\text{Fraction Power}}$)

Fraction of Max Specified SPEED		0.60	0.65	0.70	0.75	0.80	0.85	0.90	0.92	0.94	0.96	0.98	1.00
Fraction of Max Specified POWER		.216	.275	.343	.432	.512	.614	.729	.779	.831	.885	.941	1.00
Efficiency Factor	Single Flow	.774	.815	.854	.890	.923	.951	.975	.983	.989	.994	.998	1.00
	Double Flow	.766	.807	.846	.882	.915	.943	.965	.973	.979	.984	.988	.990

1. This table is valid for 10 percent efficiency factors are based on the locus of best valve points.
degree change in initial temperature.

TURBINES WITH IMPROVED LIGHT-LOAD PERFORMANCE

The basic method determines performance for straight thru turbines designed for endurance operation at 75 to 100 percent power. Performance at the lower powers and speeds can be significantly improved with some penalty in performance at the higher powers and speeds by increasing the turbine wheel speed and by adding a cruising section. The cruising section is inactivated at the higher powers by bypassing or by paralleling. Such turbines are designed specifically for a given application and performance will vary, depending on the evaluations at the specified points of operation. Guaranteed performance on such designs can be obtained by factory inquiry. However, in order to indicate the degree of tilting of performance possible the following load factors substituted for Table C of the method will give reasonable estimating performance for designs with cruising points at less than 25 percent load.

TABLE C2

Load Correction Factors

(Interpolate for Intermediate Speeds)

Fraction of Max Specified SPEED		0.45	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	0.95	1.00
Fraction of Max Specified POWER		.091	.125	.166	.216	.275	.343	.422	.512	.614	.729	.857	1.00
Efficiency Factor	2 row/1 row open close valve gear	.683	.747	.801	.843	.876	.903	.924	.941	.956	.967	.976	.985
	Series Parallel	.737	.795	.842	.877	.888	.895	.904	.914	.926	.941	.956	.973
	Interstage Bypass	.737	.795	.842	.877	.898	.909	.916	.920	.927	.935	.946	.958

TABLE D

Factors for Astern Rotation Loss

Back Pressure, Inches Hg	0.5	1.0	1.5	2.0	2.5	3.0	4.0	5.0
Factor	.998	.997	.995	.993	.992	.990	.987	.983

Steam Rates

Single-Cylinder or Cross-Compound

TABLE E

Excess Exhaust Loss (E.E.L.), BTU/Lb. (Interpolate for Intermediate Exhaust Flows and Speeds)

Exhaust Flow, lb./Hr. Exh. Press., Inches Hg X Annulus Area, ft ²	Percent Speed					
	40	50	60	70	80	100
	Excess Exhaust Loss, BTU/lb					
6000				26.6	26.8	25.8
7500				24.6	24.6	23.6
7000				24.2	22.3	21.5
6500			24.3	21.7	19.9	19.1
6000			21.9	19.3	17.6	16.8
5500			19.3	16.7	15.1	14.5
5000		19.9	16.5	14.1	12.7	12.2
4500		16.8	13.6	11.3	10.3	10.2
4000	17.6	13.7	10.7	8.8	8.1	8.4
3500	14.0	10.4	7.8	6.3	6.2	7.0
3000	10.3	7.0	5.1	4.3	4.7	6.1
2500	6.1	3.8	2.7	2.8	3.9	6.0
2000	2.3	1.2	1.1	2.2	4.4	7.6
1500	-1.0	-0.5	1.0	3.6	7.3	11.8
1300	-1.9	-0.7	1.7	4.9	9.3	14.3
1100	-2.4	-0.7	3.1	7.0	12.1	17.9
900	-2.4	0.8	5.0	9.7	15.8	

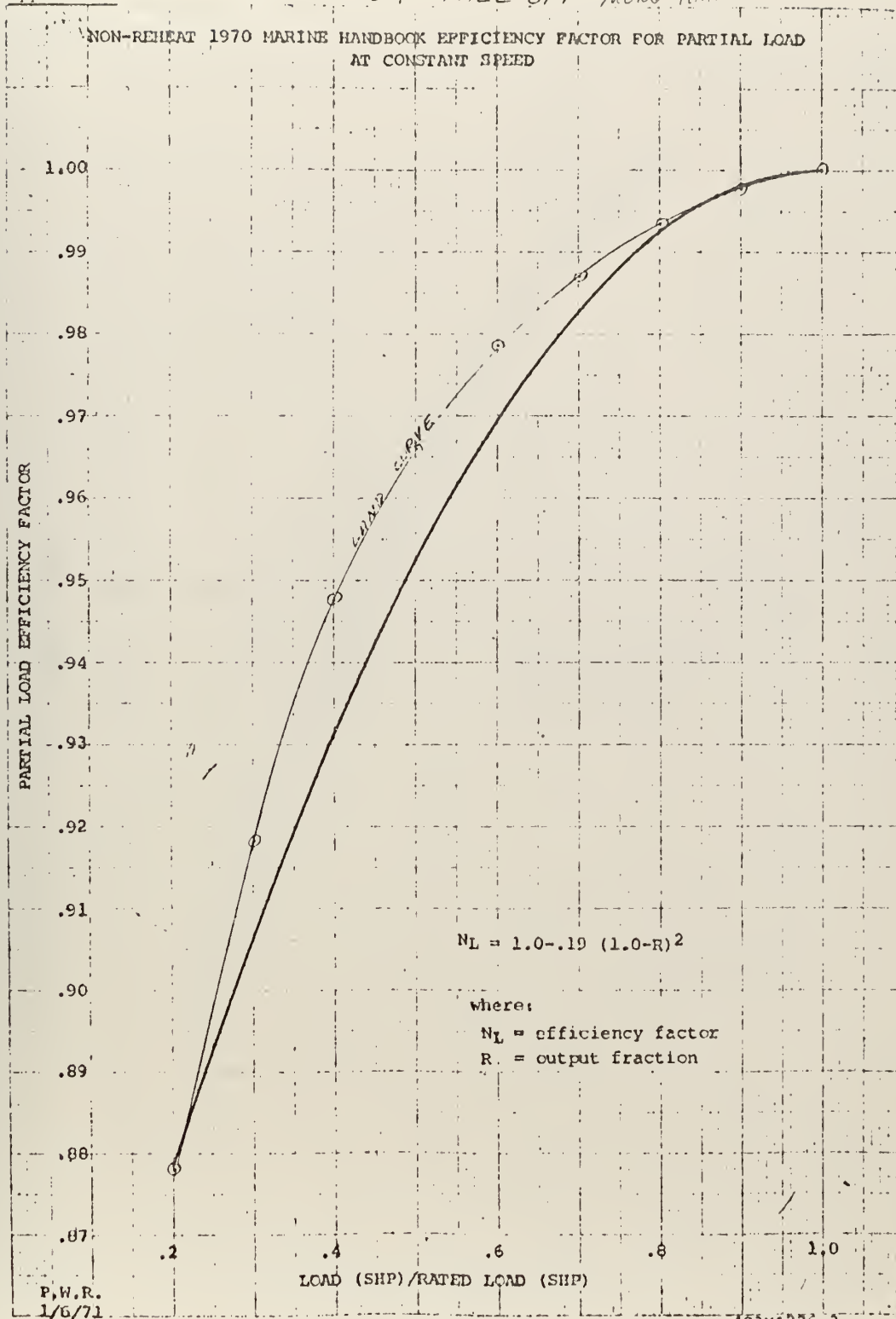
1. Excess exhaust loss is equal to used energy end point (U.E.L.P.) minus the expansion line end point (E.L.E.P.). A negative value of E.E.L. means that the internal efficiency is better than that indicated by the E.L.E.P.
2. Method Exhaust Flow = $\text{SHP} \times \text{T.S.R. (lb./hp-hr)}$
(Table A X Table B X Table C X Table D X 0.980)
3. E.L. Factor = $1.00 - \frac{1}{100} \times \frac{\text{E.E.L.}}{26.5} \times \text{T.S.R. (lb./hp-hr)}$
(Table A X Table B X Table C)
4. Available Annulus Areas (ft²)
Single-casing, Single-flow 2.40 4.99 6.76
Cross-compound, Single-flow 9.17 13.65 18.20
Cross-compound, Double-flow 24.96 30.0
5. For good practice Exhaust Flow, lb./Hr. @ 100% Speed should not be greater than 6000 nor less than 4000.
Exh. Press., inches Hg X Annulus Area, ft²
6. However those cases where Exhaust Flow, lb./Hr. @ 100% Speed is greater than 10,000
Annulus Area
lb./hr./ft² should be referred to the factory for investigation of mechanical strength.

FOR VARIABLE PRESSURE - CONST. AREA

THIS WILL NOT FALL OFF MORE THAN 11%

Appendix C

NON-REHEAT 1970 MARINE HANDBOOK EFFICIENCY FACTOR FOR PARTIAL LOAD
AT CONSTANT SPEED



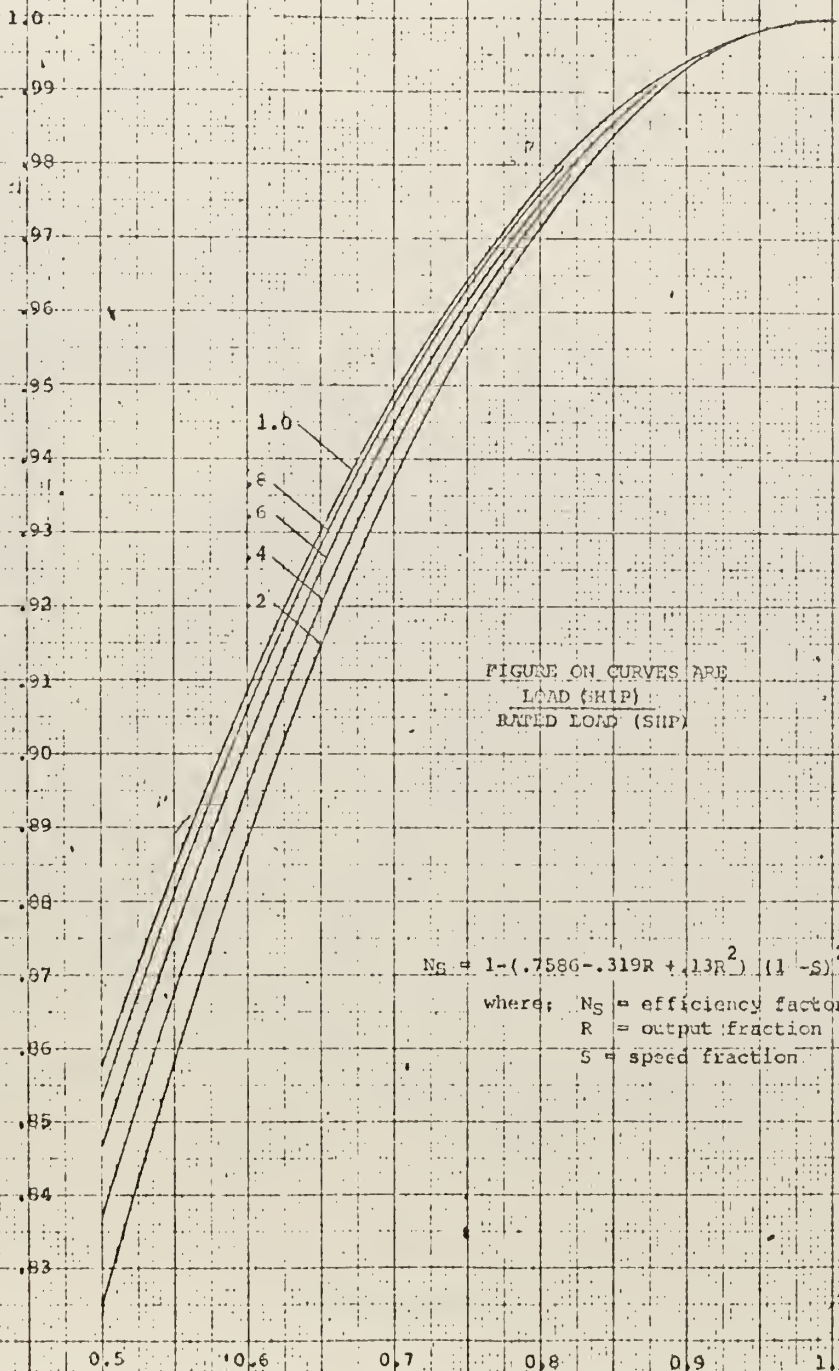
P, W, R.
1/6/71

FOR REVERSE PRESSURE BACK THIS FACTOR OUT

Appendix C

NON-REHEAT 1970 MARINE HANDBOOK EFFICIENCY FACTOR FOR REDUCED SPEED

SPEED EFFICIENCY FACTOR



P.W.R.
8/12/70

SPEED (RPM)/RATED SPEED (RPM)

Appendix D

SUMMARY OF BOILER CALCULATIONS

	<u>Cruise Power</u>		<u>Full Power</u>	
	<u>Temp.</u> <u>(°F)</u>	<u>Pressure</u> <u>(psig)</u>	<u>Temp.</u> <u>(°F)</u>	<u>Pressure</u> <u>(psig)</u>
Waterside:				
Economizer Inlet	280	415	280	1275
Economizer Outlet	448	415	407	1275
Drum	448	415	578	1275
Superheater Outlet	758	400	997	1200
Desuperheater Outlet	- -	- -	655	1160
Fireside:				
Firebox	950		2752	
After Superheater	816		1757	
After Main Bank	492		761	
Stack	364		355	
Flow:				
Steam, Superheated	25,000 Lb/Hr		230,000 Lb/Hr	
Steam, Desuperheated	0 Lb/Hr		30,000 Lb/Hr	
Air	194,000 Lb/Hr		293,238 Lb/Hr	
Draft Losses:				
Boiler (less burners)	4.36 in. water		20 in. water	

Appendix D

CRUISE MODE BOILER PERFORMANCE OF THE SCREEN BANK

April 5, 1972

1.	Tube Size & Thickness	2	Long Spacing	2 1/2
2.	No Rows Deep	3	Trans Spacing	4 1/4
3.	No Tubes Wide	32-31-32	In Line	(✓)
			Stgged	()
4.	Rating		Product	10.62
5.	Lb. Gas/hr W	194,000		
6.	Free Area	75.8		
7.	Mass Flow	2559		
8.	Eff Surface S	595		
9.	Gas Temp ent T_1	950		
10.	Est Gas Temp lvg	900		
11.	Avg Gas Temp	925		
12.	Cp	.268		
13.	Sat Steam Temp T_s	448		
14.	Beam Length	4.05		
15.	Rr	.94		
16.	Diam Correction	.812		
17.	Fa	.79		
18.	Rc (Curve M-)	9.75		
19.	Rc Corrected	6.249		
20.	R	7.189		
21.	Aloge RS/WCp	1.086		
22.	$T_1 - T_s$	502		
23.	$T_2 - T_s$	462		
24.	Gas Temp lvg T_2	910		
25.	Gas Temp ent Shtr			
26.	Btu/lb Gas ent Shtr			
27.	Mm Btu/hr Gas ent Shtr			
28.	Mm Btu/hr Absorbed as SM			
29.	Mm Btu/hr Gas lvg Shtr			
30.	Btu/lb Gas lvg Shtr			
31.	Gas Temp lvg Shtr			
32.	Draft Loss/Row			
33.	f	.0248		
34.	Draft Loss			

form 231

Calc. by Horlitz

Appendix D

CRUISE MODE BOILER PERFORMANCE OF THE SUPERHEATER

April 5, 1972

1.	Tube Size and Thickness	1 1/2	
2.	No Tubes Wide and Spacing	62@2 1/8	
3.	No Tubes Deep	6	
4.	Effective Length/Tube		
5.	Effective Heating Surface S	1658	
6.	Gas Free Area	46.1	
7.	Rating		
8.	Gas - Flow Wg	194,000	
9.	Temp ent T	910	
10.	Temp lvg T ₂ ¹	835	816 by formulas
11.	Temp avg	872	
12.	Specific Heat Cp	.267	
13.	Steam - Flow Ws	25,000	
14.	Temp ent t ₁	448	
15.	Temp lvg t ₂	696	758 by formulas
16.	Temp Rise	248	
17.	Enthalpy Change	155	
18.	Specific Heat Cs	.625	
19.	Gas Mass Flow G	4208	
20.	Transfer Rate R	15	
21.	RS/ WgCp		
22.	WgCp/ WsCs	3.31	
23.	(T ₁ - T ₂) / (T ₁ - t ₁)		
24.	T ₁ - t ₁		
25.	T ₁ - T ₂		
26.	T ₂		
27.	WgCp(T ₁ - T ₂) / Ws		
28.	Btu/lb Stm ent		
29.	Btu/lb Stm lbg		
30.	Shtd Stm Temp		
31.	No Steam Passes		
32.	Avg Specific Volume v		
33.	Actual Element Length		
34.	Equiv Length Bends, etc.		
35.	Total Equiv Length L		
36.	Lb Shtd Stm/min/Element F		
37.	Tube Inside Diameter D		
38.	F ^{1.85}		
39.	D ^{4.97}		
40.	Shtr. Pressure Drop		

form 324

Calc. by Horlitz

Appendix D

CRUISE MODE BOILER PERFORMANCE OF THE MAIN BANK

April 5, 1972

1.	Tube Size & Thickness	1	Long Spacing	2
2.	No Rows Deep	28	Trans Spacing	1 17/32
3.	No Tubes Wide	91	In Line	(✓)
			Stg'd	()
4.	Rating		Product	3.0625
5.	Lb. Gas/hr W	194,000		
6.	Free Area	48.7		
7.	Mass Flow	3984		
8.	Eff Surface S	7812		
9.	Gas Temp ent T_1	816		
10.	Est Gas Temp lvg	475		
11.	Avg Gas Temp	646		
12.	Cp	.261		
13.	Sal Steam Temp T_s	448		
14.	Beam Length	2.45		
15.	Rr	.46		
16.	Diam Correction	1.00		
17.	Fa	1.022		
18.	Rc (Curve M-49B)	13.0		
19.	Rc Corrected	13.286		
20.	R	13.746		
21.	Aloge RS/WCp	8.34		
22.	$T_1 - T_s$	368		
23.	$T_2 - T_s$	44		
24.	Gas Temp lvg T_2	492		
	RN = 5100			
25.	Gas Temp ent Shtr			
26.	Btu/lb Gas ent Shtr			
27.	Mm Btu/hr Gas ent Shtr			
28.	Mm Btu/hr Absorbed as SM			
29.	Mm Btu/hr Gas lvg Shtr			
30.	Btu/lb Gas lvg Shtr			
31.	Gas Temp lvg Shtr			
32.	Draft Loss/Row	.31		
33.	f	.038		
34.	Draft Loss 38 Tubes	.42"		

Appendix D

CRUISE MODE PRELIMINARY HEAT BALANCE OF THE BOILER

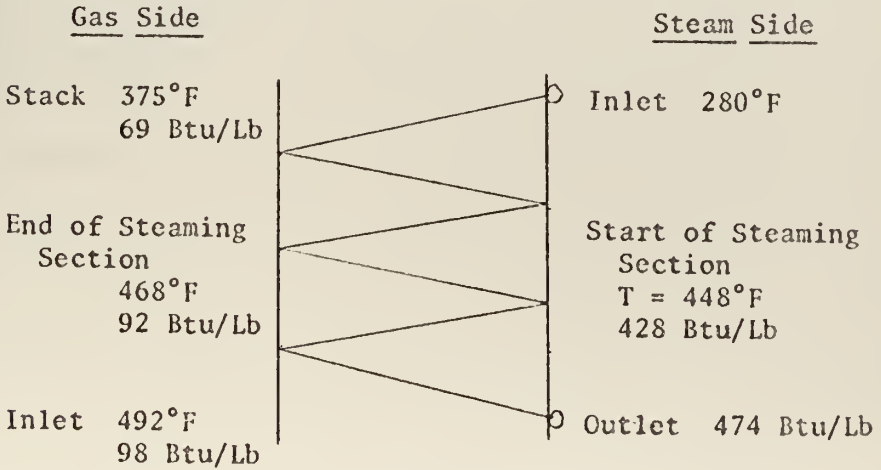
Steam Side

Input Conditions: Water at 280°F = 249 Btu/Lb
Output Conditions Steam at 400 psig and 758°F = 1394 Btu/Lb
Flow Rate: 25000 Lb/Hr
Heat Gain: (25000) Lb/Hr (1394 - 249) Btu/Lb = 28,625,000 Btu/Hr

Air Side

Input Conditions: 950°F = 219 Btu/Lb
Flow Rate: 194,000 Lb/Hr
Input Heat: (194,000) Lb/Hr (219) Btu/Lb = 42,486,000 Btu/Hr
Less Steam Side Heat Gain -28,625,000 Btu/Hr
Output Heat: 13,861,000
Output Conditions: 71 Btu/Lb and 384°F Stack Temperature
Goal is Stack Temperature of 375°F so Heat Gain to Steam Side
would be: (194,000) Lb/Hr (0.261) Btu/Lb°F (575)°F = 29,114,550 Btu/Hr

CRUISE MODE ECONOMIZER HEAT BALANCE



Appendix D

CRUISE MODE BOILER PERFORMANCE OF THE

ECONOMIZER STEAMING SECTION

1.	Tube Size & Thickness	
2.	No Rows Deep	
3.	No Tubes Wide	
	S.S. 15 Wide x 11.67' Bts 8 Rows High	
4.	Rating	
5.	Lb. Gas/Hr W	194,000
6.	Free Area (15)(11.67)(.134)	23.5
7.	Mass Flow	8255
8.	Eff Surface S	4667
9.	Gas Temp ent T_1	492
10.	Est Gas Temp lvg	468
11.	Avg Gas Temp	480
12.	Cp	.255
13.	Sat Steam Temp T_s	448
14.	Beam Length	
15.	Rr	
16.	Diam Correction	
17.	Fa	
18.	Rc (curve M-)	
19.	Rc Corrected	
20.	R	8.9
21.	Aloge RS/WCp .789	2.20
22.	$T_1 - T_s$	44
23.	$T_2 - T_s$	20
24.	Gas Temp lvg T_2	468
25.	Gas Temp ent Shtr	
26.	Btu/lb Gas ent Shtr	
27.	Mm Btu/hr Gas ent Shtr	
28.	Mm Btu/hr Absorbed as SM	
29.	Mm Btu/hr Gas lvg Shtr	
30.	Btu/lb Gas lvg Shtr	
31.	Gas Temp lbg Shtr	
32.	Draft Loss/Row	
33.	f	
34.	Draft loss	

Appendix D

CRUISE MODE BOILER PERFORMANCE OF THE MODIFIED

ECONOMIZER STEAMING SECTION

1.	Tube Size & Thickness	
2.	No Rows Deep	
3.	No Tubes Wide	
	S.S. 17 wide x 13' Bts 7 High	
4.	Rating	
5.	Lb. Gas/hr W	194,000
6.	Free Area 17 x 13 x .134	29.6
7.	Mass Flow	6554
8.	Eff Surface S	
9.	Gas Temp ent T_1	492
10.	Est Gas Temp lvg	468
11.	Avg Gas Temp	480
12.	Cp	.255
13.	Sat Steam Temp T_s	448
14.	Beam Length	
15.	Rr	
16.	Diam Correction	
17.	Fa	
18.	Rc (Curve M-)	
19.	Rc Corrected	
20.	R	7.9
21.	Aloge RS/WCp	2.20
22.	$T_1 - T_s$	44
23.	$T_2 - T_s$	20
24.	Gas Temp lvg T_2	468
25.	Gas Temp ent Shtr	
26.	Btu/lb Gas ent Shtr	
27.	Mm Btu/hr Gas ent Shtr	
28.	Mm Btu/hr Absorbed as SM	
29.	Mm Btu/hr Gas lvg Shtr	
30.	Btu/lb Gas lvg Shtr	
31.	Gas Temp lvg Shtr	
32.	Draft Loss/Row	
33.	f	
34.	Draft Loss	

form 231

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4-5-72

Appendix D

CRUISE MODE BOILER PERFORMANCE OF THE

NON-STEAMING ECONOMIZER

	<u>Standard Economizer</u>	<u>Modified Economizer</u>
1. Tube Size and Thickness	2" S.S.	
2. No Rows High	24	20
3. No Tubes Wide	15	17
4. Length between Tube Sheets	11.67	13.0'
5. Heating Surface/Tube		
6. Total Heating Surface S	calc. by formula 8893	10100
7. Rating		
8. Lb Gas/hr Wg	194,000	
9. Lb Water/hr Ww	25,000	
10. Gas Temp ent T_1	468	
11. Est Gas Temp lvg	375	
12. Avg Gas Temp	422	
13. Water Temp ent t_1	280	
14. Est Water Temp lvg	448	
15. Avg Water Temp	364	
16. Free Area (15)(11.67)(.134)	23.5	
17. Mass Flow	8255	
18. Transfer Rate R	10.1	8.9
19. Cp	.254	
20. Cw	1.053	
21. RS/Wg Cp		
22. WgCp / Ww Cw	1.871	
23. $(T_1 - T_2) / (T_1 - t_1)$		
24. $T_1 - t_1$		
25. $T_1 - T_2$		
26. $t_2 - t_1$		
27. Water Temp lvg Econ t_2		
28. Gas Temp lvg Econ T_2	364	
29. Avg Specific Volume		
30. Lbs Water/Element/hr		
31. Water Velocity		
32. Length/Element		
33. Econ Pressure Drop		
34. Draft Loss/Ten Tubes		
35. Total Draft Loss (In. Water)	7.56	3.94

form 232

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4-5-72

Appendix D

FIRIED MODE BOILER PERFORMANCE OF THE FURNACE (Navy Distillate Fuel Oil Fired)

	<u>Full Power</u>	<u>Overload</u>
1. Rating %	100	120
2. Total lb Steam/hr	230,000	276,000
3. Lb Dshtd Steam/hr	30,000	50,000
4. Pressure at S.O.	1200	1165
5. Total Steam Temp	950	925
6. Feedwater Temp	280	280
7. Pressure at D.S.O.	1160	1025
8. Temp at D.S.O.	655	680
9. Assumed Drum Pressure	1275	1275
10. Sat Steam Temp	578	578
11. Btu/lb Shtd Steam	1470	1457
12. Btu/lb Dshtd Steam	1278	1309
13. Btu/lb Sat Steam	1180	1180
14. Btu/lb Feedwater	249	249
15. Btu/lb Feedwater to Sat Steam	931	931
16. Btu/lb Sat to Shtd Steam	290	277
17. Btu/lb Shtd to Dshtd Steam	192	148
18. Mm Btu/hr Feed to Sat Steam	214.130	256.956
19. Mm Btu/hr Sat to Shtd Steam	66.700	76.452
20. Mm Btu/hr Shtd to Dshtd Steam	-5.760	-7.400
21. Mm Btu/hr Total	275.070	326.008
15% x-s Air A/F Ratio =	16.9	16.9
22. Efficiency	86.1	85.7
23. Gross H V of Fuel NDFO	19,300	19,300
24. Lb Fuel/hr	16,382	19,641
25. Lb Air/hr	276,856	331,933
26. Lb Gas/hr	293,238	351,574
27. Furnace Volume	1130	
28. Btu Released/cu ft F V	279,798	335,461
29. Air Temp at Burners	100	100
30. Net H V of Fuel	17,975	17,975
31. Effective R H S	422	
32. Btu Available/sq ft R H S	697,788	836,604
33. Btu Not Absorbed/sq ft R H S	523,000	639,000
34. Btu Absorbed/sq ft R H S	174,788	197,604
35. Btu/lb Gas	753	767
36. Gas Temp lvg Furnace	2752	2796

Appendix D

FIRED MODE BOILER PERFORMANCE OF THE SCREEN BANK

April 5, 1972

1.	Tube Size & Thickness	2	O.D. x O. 134 MWT	In Line (✓)
2.	No Rows Deep	3	Long Spacing 2 1/2	Stgged ()
3.	No Tubes Wide	32-31-32	Trans Spacing 4 1/4	Prod. 10.62
4.	Rating %	100		120
5.	Lb. Gas/hr W	293,238		351,574
6.	Free Area	75.8		
7.	Mass Flow	3869		4638
8.	Eff Surface S	595		
9.	Gas Temp ent T_1	2752		2796
10.	Est Gas Temp lvg	2580		2636
11.	Avg Gas Temp	2666		2716
12.	Cp	.318		.319
13.	Sat Steam Temp T_s	578		578
14.	Beam Length	4.02		
15.	Rr	3.08		3.12
16.	Diam Correction	.812		.812
17.	Pa	.795		.813
18.	Rc (Curve M-49B)	15.27		17.3
19.	Rc Corrected	9.85		11.42
20.	R	12.93		14.54
21.	Aloge RS/WCp	1.088		1.080
22.	$T_1 - T_s$	2174		2218
23.	$T_2 - T_s$	1998		2054
24.	Cas Temp lvg T_2	2576		2632
25.	Gas Temp ent Shtr	2576		2632
26.	Btu/lb Gas ent Shtr	697		714
27.	Mm Btu/hr Gas ent Shtr	204.387		251.024
28.	Mm Btu/hr Absorbed as SM	66.700		76.45
29.	Mm Btu/hr Gas lvg Shtr	137.687		174.574
30.	Btu/lb Gas lvg Shtr	470		497
31.	Gas Temp lvg Shtr	1840		1931
32.	Draft Loss/Row			
33.	f			
34.	Draft Loss			

Appendix D

FIRED MODE BOILER PERFORMANCE OF THE SUPERHEATER

April 5, 1972

1.	Tube Size and Thickness	1 1/2	
2.	No Tubes Wide and Spacing	62@2 1/8	
3.	No Tubes Deep	8	
4.	Effective Length/Tube		
5.	Effective Heating Surface S	2210	
6.	Gas Free Area	46.1	
7.	Rating %	100	120
8.	Gas - Flow Wg	293,238	351,574
9.	Temp ent T_1	2576	2632
10.	Temp lvg T_2	1840	1931
11.	Temp avg	2208	2282
12.	Specific Heat Cp	.305	.307
13.	Steam - Flow Ws	230,000	276,000
14.	Temp ent t_1	578	578
15.	Temp lvg t_2	950	925
16.	Temp Rise	372	347
17.	Enthalpy Change	290	277
18.	Specific Heat Cs	.780	.798
19.	Gas Mass Flow G	6361	7626
20.	Transfer Rate R	25.0	28.3
21.	RS / WgCs	.617	.579
22.	WgCp / WsCs	.499	.490
23.	$(T_1 - T_2) / (T_1 - t_1)$.410	.394
24.	$T_1 - t_1$	1998	2054
25.	$T_1 - T_2$	819	809
26.	T_2	1757	1823
27.	$WgCp(T_1 - T_2) / Ws$	318	316
28.	Btu/lb Stm ent	1180	1180
29.	Btu/lb Stm lvg	1498	1496
30.	Shtd Stm Temp	997	992
31.	No Steam Passes		
32.	Avg Specific Volume v		
33.	Actual Element Length		
34.	Equiv Length Bends, etc.		
35.	Total Equiv Length L		
36.	Lb Shtd Stm/min/Element F		
37.	Tube Inside Diameter D		
38.	F1.85		
39.	D4.97		
40.	Shtr. Pressure Drop		

form 324

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Appendix D

FIRED MODE BOILER PERFORMANCE OF THE MAIN BANK

April 5, 1972

1.	Tube Size & Thickness	1	Long Spacing 2	In Line (✓)
2.	No Rows Deep	28	Trans Spacing 1	17/32
3.	No Tubes Wide	91	Product	3.0625
4.	Rating %	100		120
5.	Lb Gas/hr W	293,238		351,574
6.	Free Area	48.7		
7.	Mass Flow	6021		7219
8.	Eff Surface S	7812		
9.	Gas Temp ent T_1	1840		1931
10.	Est Gas Temp lvg	690		730
11.	Avg Gas Temp	1265		1330
12.	Cp	.279		.281
13.	Sat Steam Temp T_s	578		578
14.	Beam Length	2.42		
15.	Rr	1.09		1.15
16.	Diam Correction	1.0		1.0
17.	Fa	1.013		1.011
18.	Rc (Curve M-49B)	18.9		21.1
19.	Rc Corrected	19.14		21.33
20.	R	20.24		22.48
21.	Aloge RS/WCp	6.91		5.92
22.	$T_1 - T_s$	1262		1353
23.	$T_2 - T_s$	183		229
24.	Gas Temp lvg T_2	761		807
25.	Gas Temp ent Shtr			
26.	Btu/lb Gas ent Shtr			
27.	Mm Btu/hr Gas ent Shtr			
28.	Mm Btu/hr Absorbed as SM			
29.	Mm Btu/hr Gas lvg Shtr			
30.	Btu/lb Gas lvg Shtr			
31.	Gas Temp lvg Shtr			
32.	Draft Loss/Row			
33.	f			
34.	Draft loss			

Appendix D

FIRED MODE BOILER PERFORMANCE OF THE ECONOMIZER

April 5, 1972

1. Tube Size and Thickness	2" spiral	
2. No Rows High	20	
3. No Tubes Wide	17	
4. Length between Tube Sheets	13.0	
5. Heating Surface/Tube		
6. Total Heating Surface S	14,740	
7. Rating %	100	120
8. Lb Gas/hr Wg	293,238	351,574
9. Lb Water/hr Ww	230,000	276,000
10. Gas Temp ent T_1	761	807
11. Est Gas Temp lvg	340	360
12. Avg Gas Temp	551	583
13. Water Temp ent t_1	280	280
14. Est Water Temp lvg	450	460
15. Avg Water Temp	365	370
16. Free Area (17)(13)(.134)	29.6	
17. Mass Flow	9907	11,878
18. Transfer Rate R	11.6	12.8
19. Cp	.258	.258
20. Cw	1.054	1.057
21. RS/Wg Cp	2.260	2.080
22. WgCp / Ww Cw	.312	.311
23. $(T_1 - T_2) / (T_1 - t_1)$.845	.823
24. $T_1 - t_1$	481	527
25. $T_1 - T_2$	406	434
26. $t_2 - t_1$	127	135
27. Water Temp lvg Econ t_2	407	415
28. Gas Temp lvg Econ T_2	355	373
29. Avg Specific Volume		
30. Lbs Water/Element/hr		
31. Water Velocity		
32. Length/Element		
33. Econ Pressure Drop		
34. Draft Loss/Ten Tubes	5.0	7.4
35. Total Draft Loss	10.0	14.8

form 232

Calc. by Horlitz

Appendix D

FIRED MODE BOILER HEAT BALANCE

April 5, 1972

1.	Rating	100	120
2.	Ambient Air Temp T_a	100	100
3.	Exit Gas Temp T_g	355	373
4.	Air Temp to Burners	100	100
5.	Lb Wet Gas/lb Oil	17.9	17.9
6.	Lb Wet Air/lb Oil	16.9	16.9
7.	Lb Moisture in Air/lb Oil	.279	.279
8.	Lb H_2O due to H_2 in Oil/lb Oil	1.260	1.260
9.	Lb Dry Gas/lb Oil	16.361	16.361
10.	H H V of Oil	19,300	19,300
11.	$T_g - T_a$	255	273
12.	$(0.46)T_g$	163	172
13.	$(1089) + (0.46)T_g - T_a$ Losses	1152	1161
14.	Due to Moisture in Air	.173	.185
15.	Due to Moisture from H_2 in Oil	7.521	7.580
16.	Dry Gas	5.187	5.553
17.	Total Calculated	12.881	13.318
18.	Rad and Unaccounted for	1.000	1.000
19.	Total	13.881	14.318
20.	Efficiency by Difference	86.119 86.1	85.682 85.7

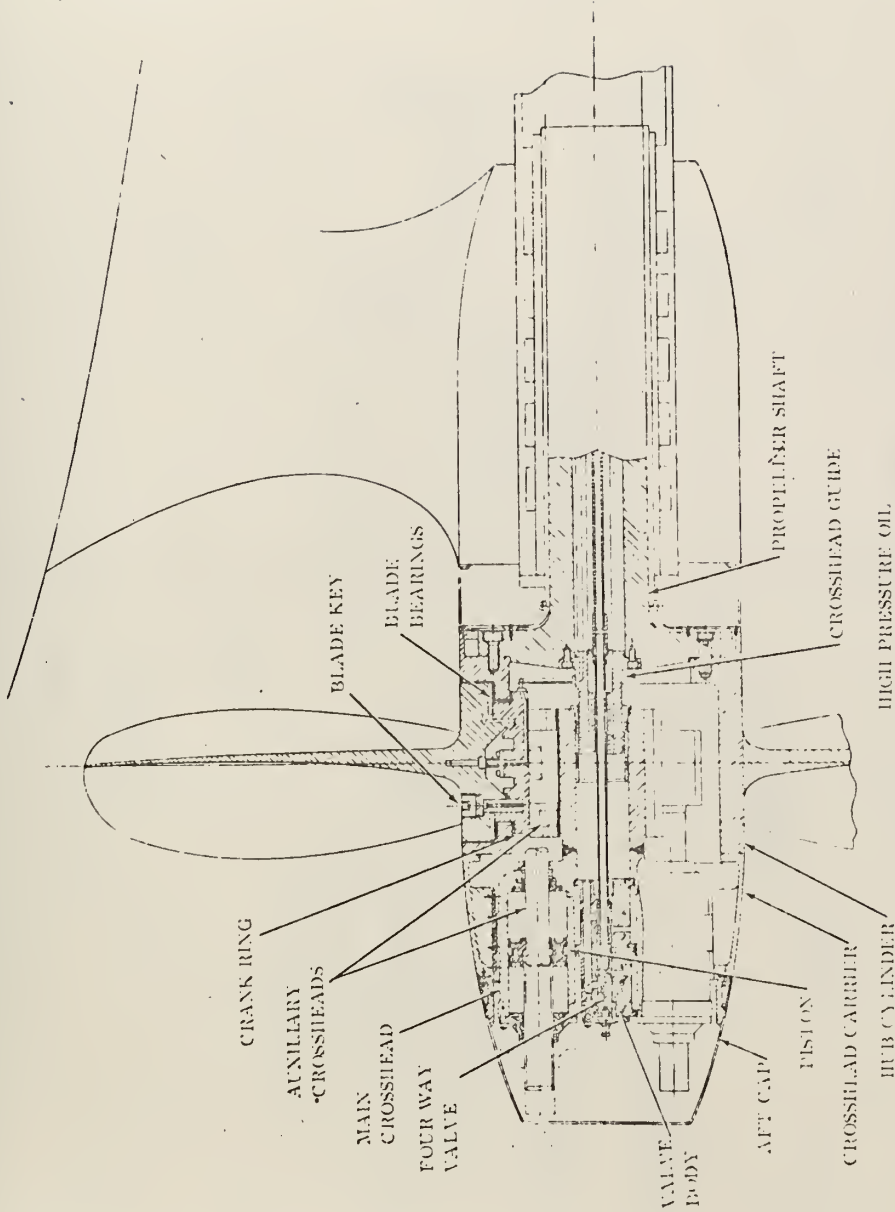


Figure 1. Hub and Blade Assembly

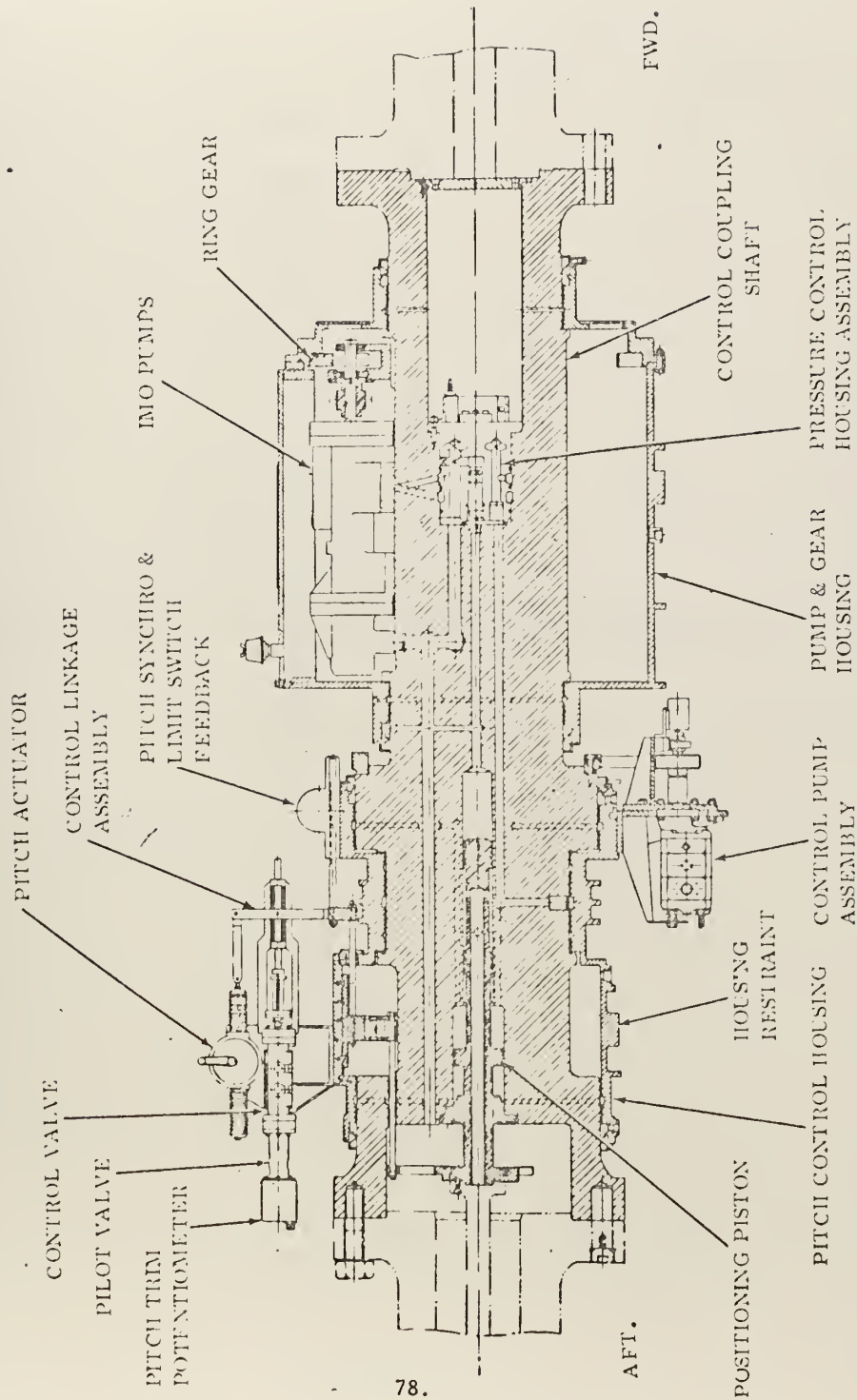
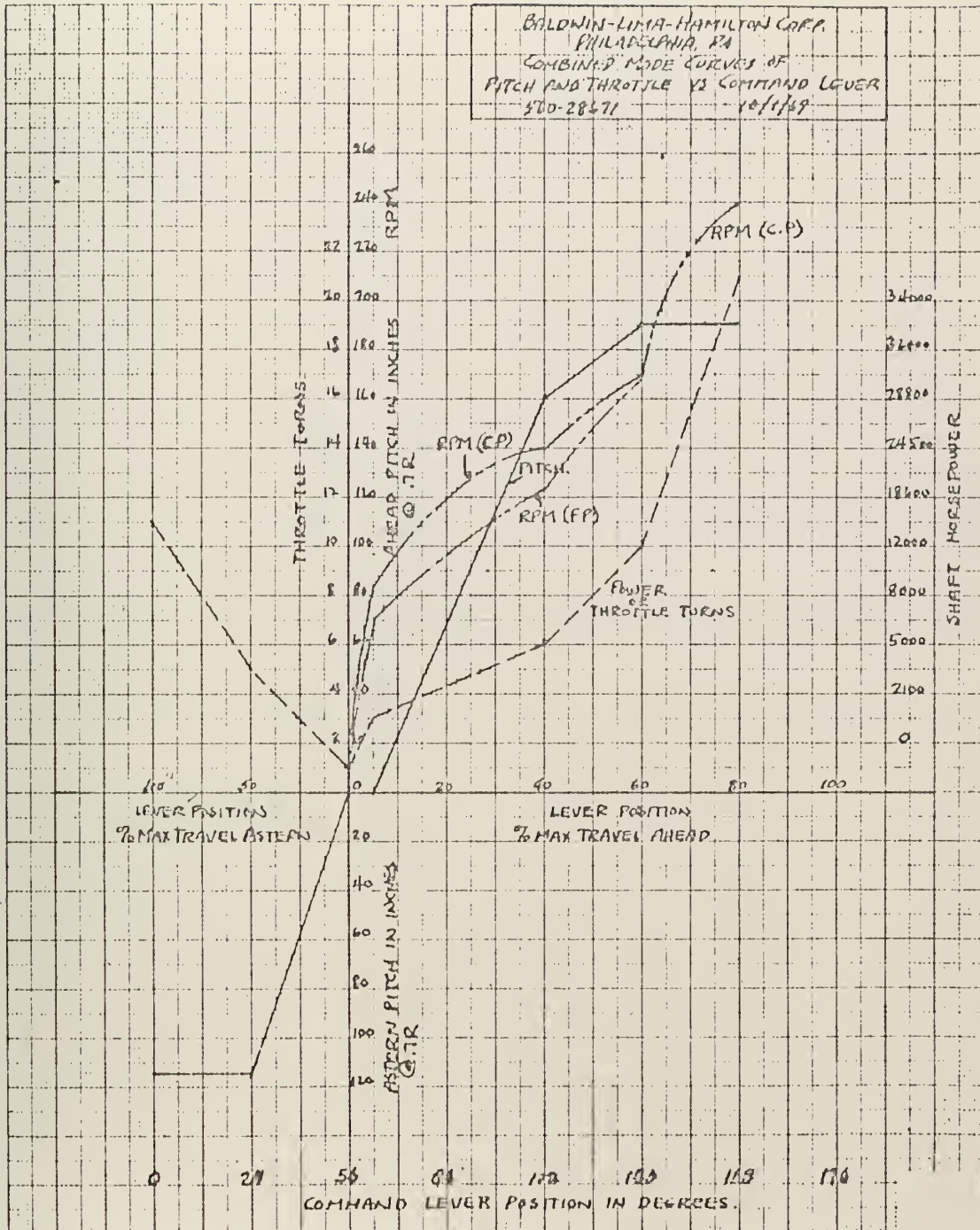


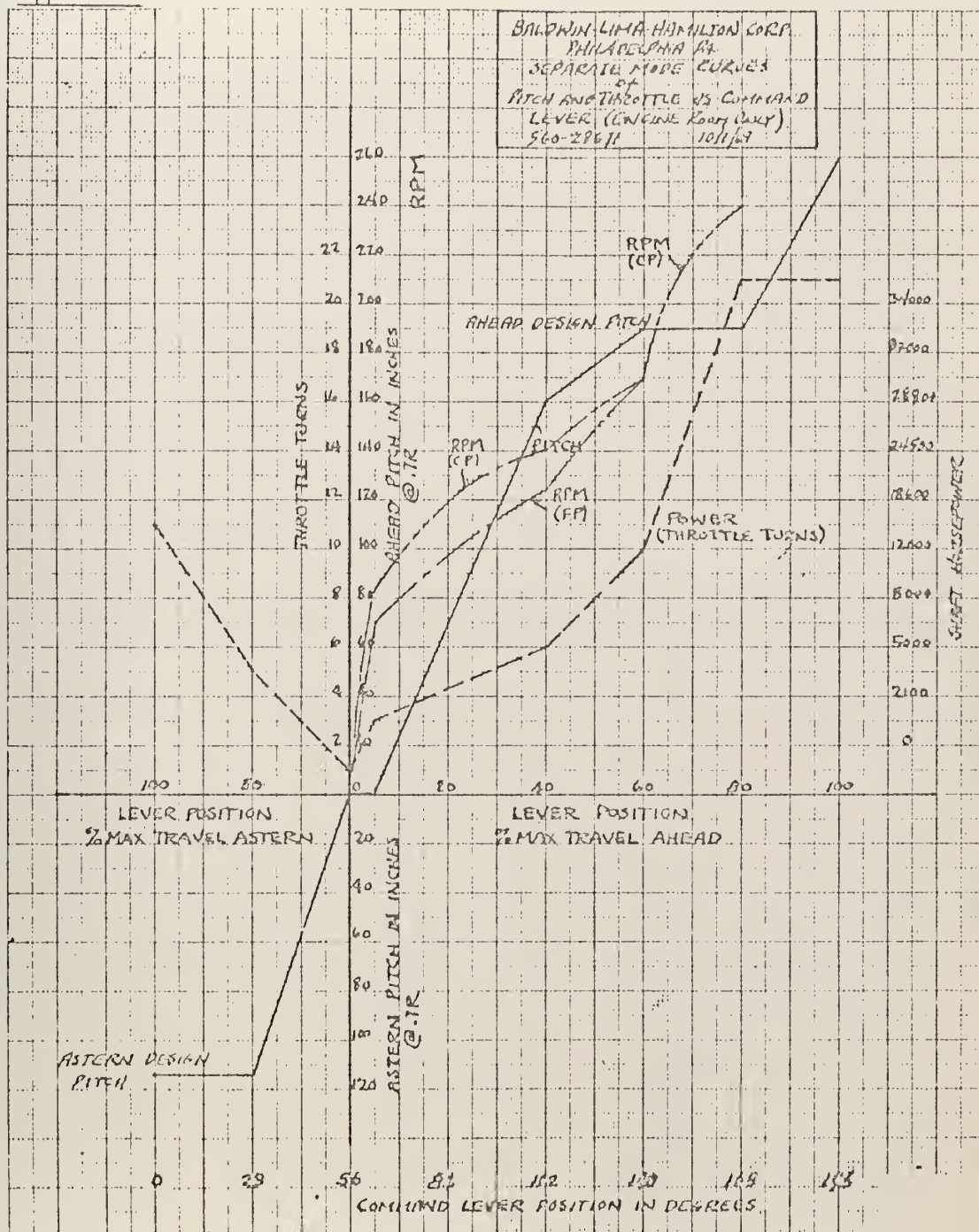
Figure 2. Control Coupling Assembly

Appendix E



CURVE A

Appendix E



CURVE B

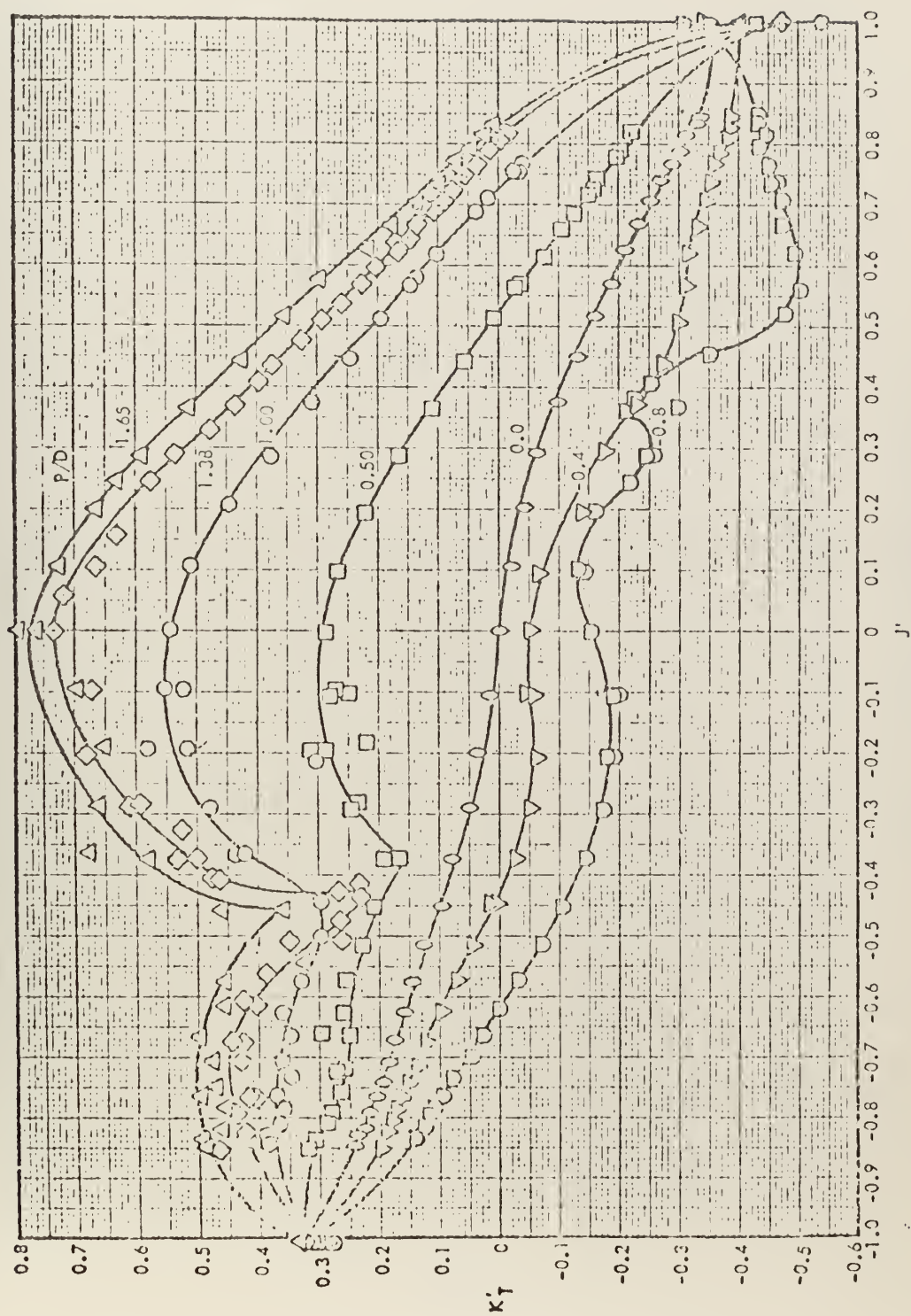


FIGURE 5 - CP PROPELLER THRUST CHARACTERISTICS.

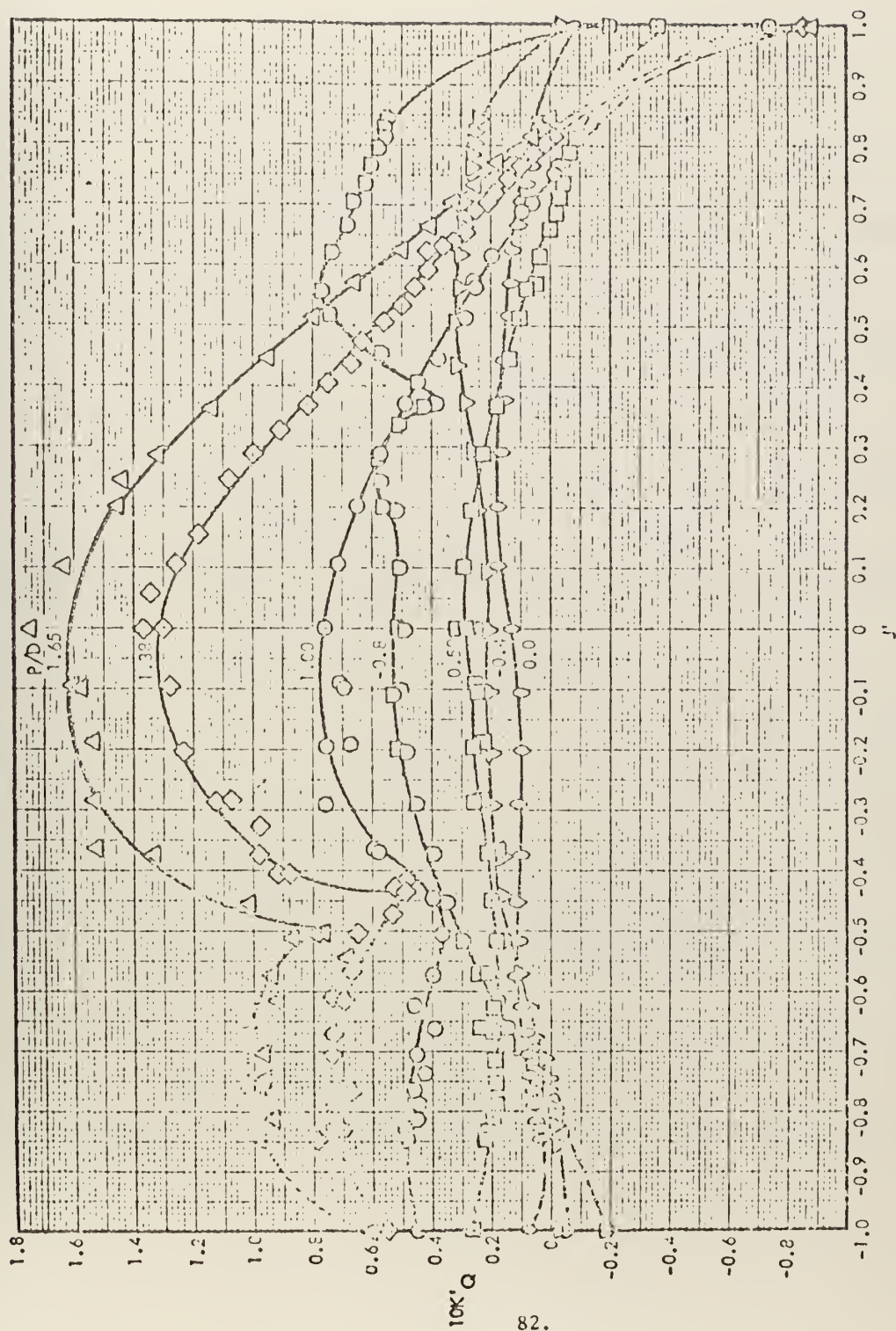


FIGURE 6 - CP PROPELLER TORQUE CHARACTERISTICS

Appendix E

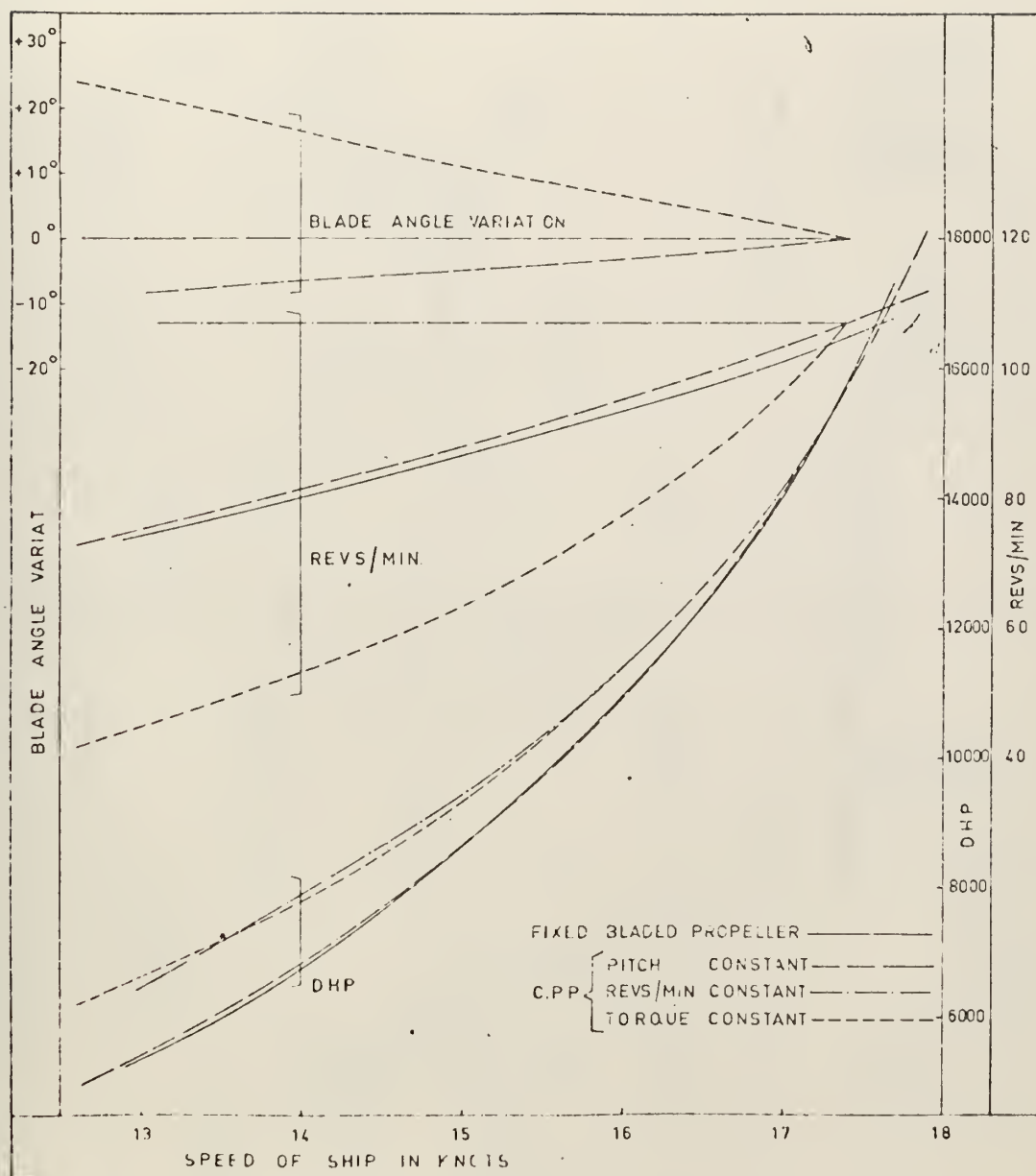
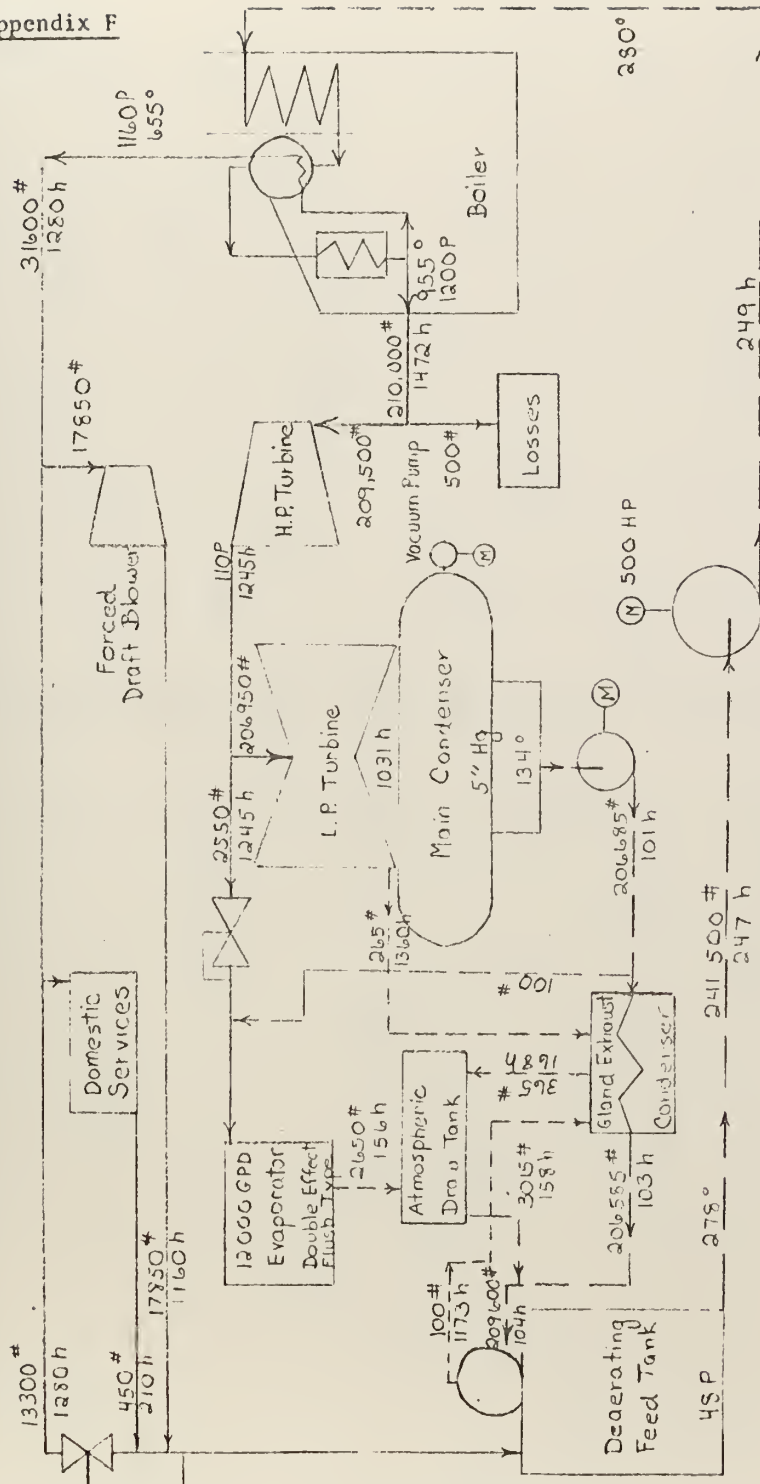


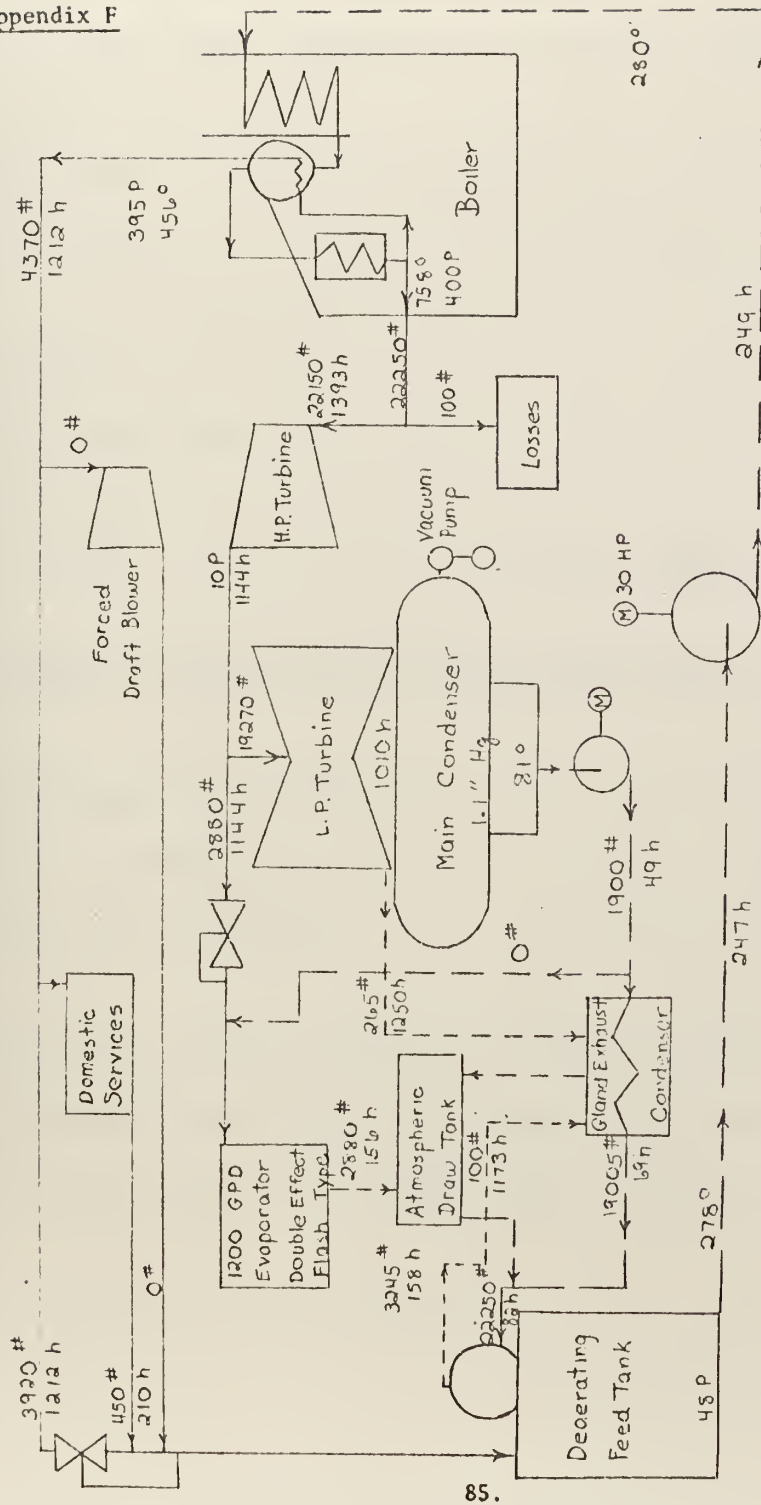
Fig. 8—Propulsion curves of Tanker when propelled by the fixed blade propeller and when propelled by the controllable pitch propeller; the latter respectively in the 0° blade setting (design setting), in settings for constant revs per min., and in settings for constant torque.

Appendix F

Operating Conditions for Total Plant

Full Power:	HP Turbine	18680	Fuel (Distillate):	Boiler	16382	Lb/Hr
	LP Turbine	17400		Gas Turbine	3402	Lb/Hr
	Gas Turbine	5820		Elec. Gen. G.T.	374	Lb/Hr
		<u>41900</u>		(750KW/2)	<u>20158</u>	Lb/Hr
	After Gear Loss (600 HP)	41300		s. f. c.	0.484	Lb/SHP-Hr
	After Attached Gen. Load (500 KW)	41615		KW = 875 per plant		

Appendix F



Operating Conditions for Total Plant

Cruise:	HP Turbine	2165 HP	Fuel (Distillate):	Boiler	0 Lb/Hr
	LP Turbine	1015 HP		Gas Turbine	3402 Lb/Hr
	Gas Turbine	5760 HP		All Purpose SFC	0.438 Lb/SHP-Hr
		8940 HP			
	After Gear Loss (500 HP)	8440 HP			
	After Taking Gen. Load (500 Kw)	7755 SHP			
					KW = 500 per plant

Appendix F

CRUISE POWER CALCULATIONS (based on Sheldon)

1. Percent exhaust heat theoretically available to Rankine Cycle

$$A = 85.5 \%$$

$$B = 0.95$$

$$A \times B = 81.2$$

$$C = 0.85\%$$

$$A \times B \times C = 69.0$$

2. Steam Flow

$$F = 0.149 \text{ lb/lb}$$

$$G = 0.90$$

$$E = 92\%$$

$$F \times G \times C \times \frac{E}{100} = 0.1053 \frac{\text{Lb steam}}{\text{Lb air}}$$

$$\text{Steam Flow} = \text{Gas Flow} \times 0.1053 = 20,500 \text{ Lb/Hr}$$

3. Step 1 corrected to 1.1 In. Hg

$$D = 2 \times 0.1053 = 0.211$$

$$A \times B \times C - D = 68.8$$

4. Exhaust heat actually available to the Rankine Cycle at 1.1 In. Hg

$$\text{Available Heat (Ideal)} = 225 \text{ BTU/Lb} \times 0.99 = 223 \text{ BTU/Lb}$$

$$\text{Available Heat} = 223 \times \frac{69.0}{100} \times \frac{92}{100} \times 194,000 = 27.1 \times 10^6 \text{ BTU/Hr}$$

5. Heat available to steam turbine

$$J = 36.9 \%$$

$$K = 0.998$$

$$\text{Heat Avail. to Stm. Turbine} = (27.1 \times 10^6) \times 0.369 \times 0.998 = 9.98 \times 10^6 \text{ BTU/Hr}$$

Appendix F

6. Actual steam turbine output

Engine Efficiency = 0.79

$$\text{Output} = \frac{9.98 \times 10^6 \times 0.79}{2544} = 3100 \text{ HP}$$

11 SEP 77
5 APR 79

24421
25171

Thesis
C5178 Clough

135356

A cogas propulsion
cycle with peak effi-
ciency at low power.

11 SEP 77
5 APR 79

24421
25171

Thesis
C5178 Clough

13 3356

A cogas propulsion
cycle with peak effi-
ciency at low power.

thesC5178

A cogas propulsion cycle with peak effic



3 2768 002 10321 0

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